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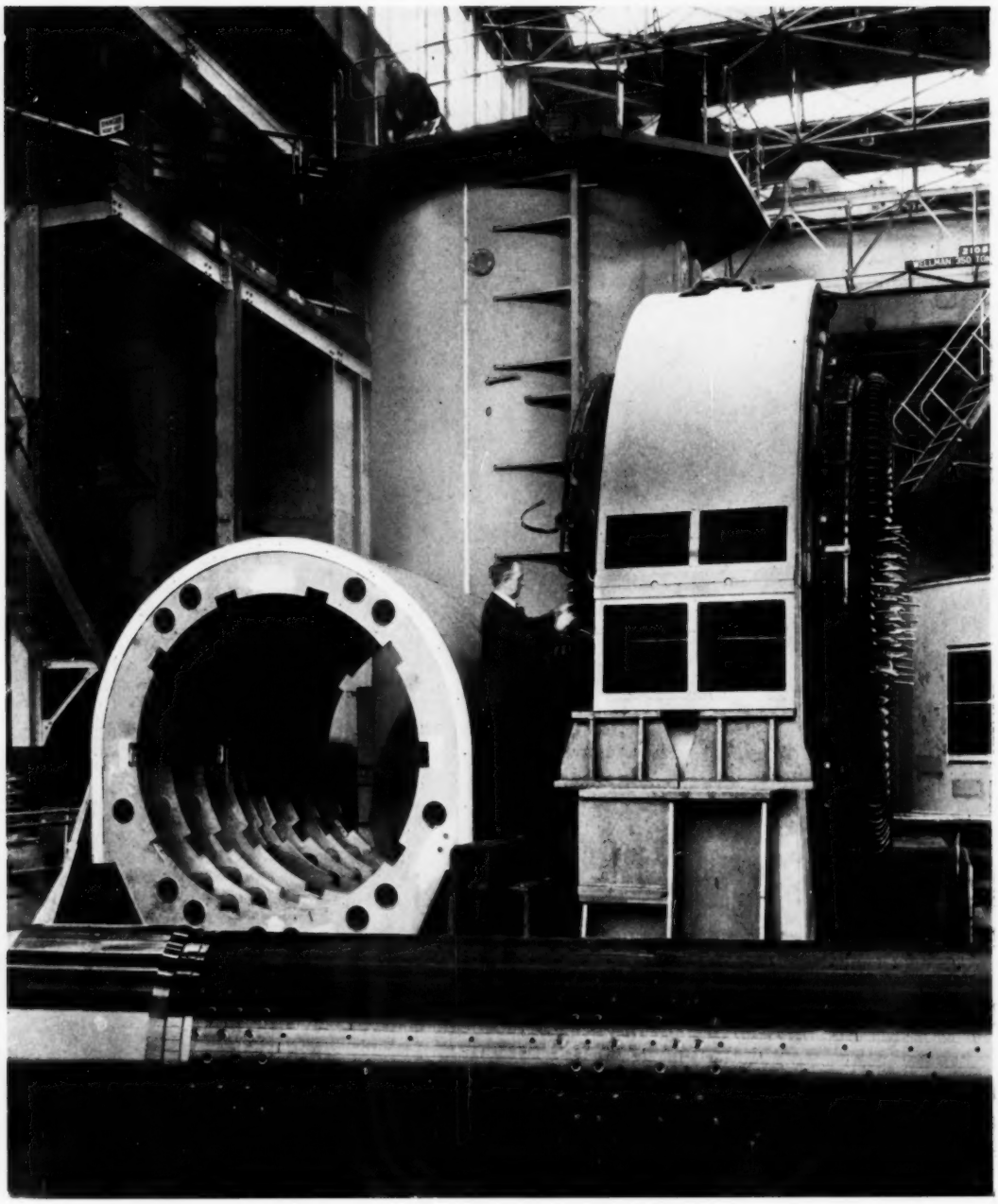


Some components of heavy electrical machines in The English Electric Company's Stafford Works

Some Problems of Large-Scale Measurement in the Heavy Electrical Industry

By T. P. JOLLY, Chief Inspector.

This article is in all essentials a reproduction of the paper presented by the author to the National Physical Laboratory for the Symposium on Engineering Dimensional Metrology in October 1953.

Many discussions have taken place during the last few years concerning the extension of the tolerance charts in the British Standard for limits and fits to include suitable tolerances to cover diameters greatly in excess of the present range, from 0·04″ to 19·686″. The maximum tolerance allowance at the upper end of the range is +0·004″ for holes in Grade IT.8 which is the most commonly used commercially.

Engineering works have consequently extended the table arbitrarily to suit their particular type of manufacture without, it is suggested, giving sufficient consideration to the overall problem of the accuracy of machining and/or measurement.

Therefore, in order to give assistance in the problem of compiling tolerance tables for large diameters, the author proposes to go in some detail into the many problems confronting the works when undertaking the manufacture of large electrical and mechanical engineering components and assemblies, and to show as far as possible within the scope of this article how the parts are measured.

Basic Standards

In order to measure comparatively large diameters of, say, 10 to 20 feet (3,050 to 6,100 mm) it is important that the *basic* standards, taking the form of length bars, have been accurately calibrated. The reference length bars and sets of 81-piece slip gauges are the most general form which the standards take, each being provided with a certificate of calibration by the National Physical Laboratory.

To maintain accurate relations of these standards to shop measuring equipment, most large engineering works are equipped with a standards room, air conditioned and temperature controlled.

Included in the equipment of the Standards

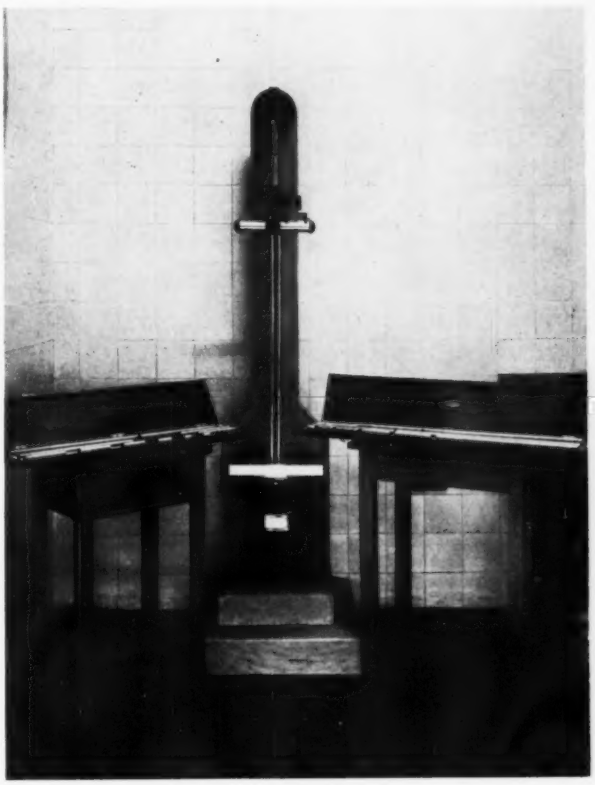


Fig. 1.—*Brooks level comparator*

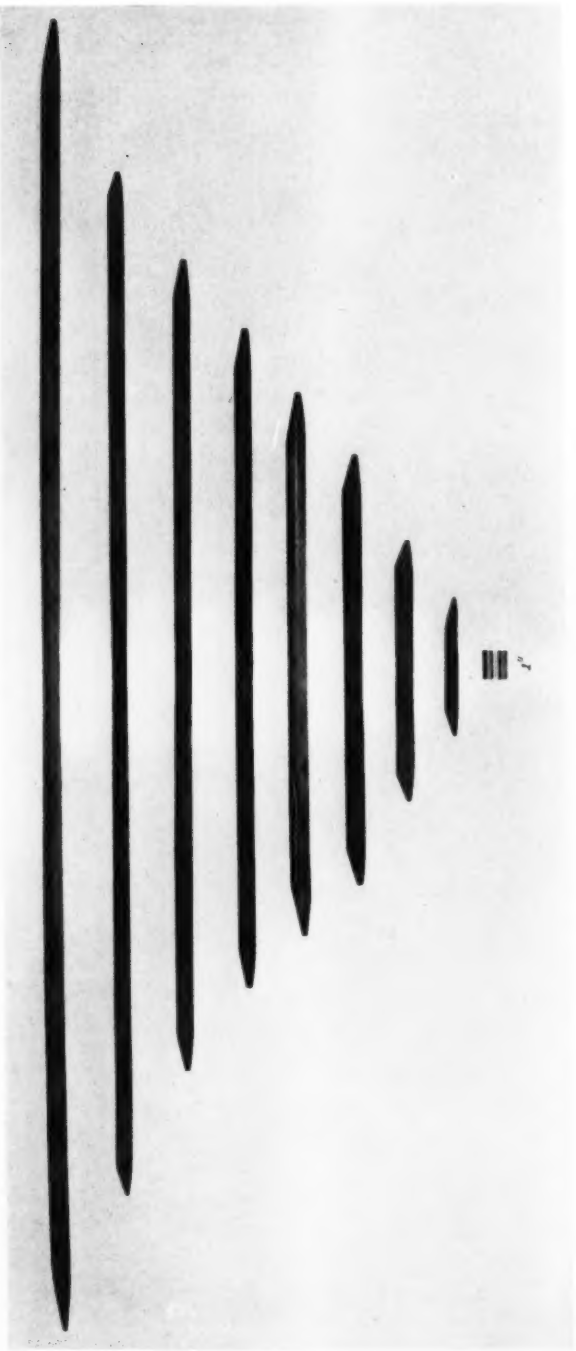


Fig. 2.—A selection of square-section point gauges

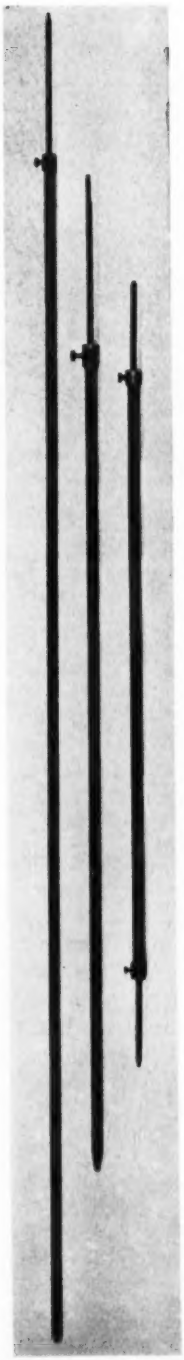


Fig. 4.—Examples of telescopic point gauges

Room at the Stafford Works of The English Electric Company is a Brooks level comparator, as shown in Fig. 1, with a set of reference length bars and workshop length bars arranged at each side ; it is used for calibrating from time to time the workshop length bars against the reference standards.

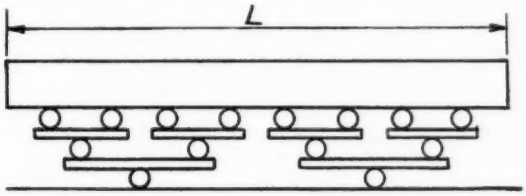


Fig. 3.—Diagrammatic arrangement of a bar supported on rollers in a compensating frame, demonstrating the location of Airy points

Shop Length Gauges

Consideration must now be given to the type of gauges used in the shops for doing the actual measurement of machined components. These gauges are called point gauges and are usually made from square section bars varying in section from $\frac{5}{16}$" (7·937 mm) for the smaller length gauges to $\frac{3}{4}$" (19·05 mm) for the longer gauges. Some examples are shown in Fig. 2. Each end of the point gauge is tapered down, the length of taper varying according to length of gauge. One side is left flat to which the taper of the other side converges ; this is done to facilitate measurements being taken near the root of a shallow diameter. The ends are made spherical, with the diameter not exceeding $\frac{1}{8}$" to $\frac{1}{4}$" (3·175 to 6·35 mm) so that the operator can get the maximum sensitivity or feel. The spherical ends are hardened and lapped to the correct radii. This type of length gauge is robust in construction, yet comparatively light to handle, and the error in length when supported at the Airy points, ·5773 of the total length apart and equi-distant from each end, can be considered negligible.

An explanation of the method of support at the Airy points referred to may be of interest here. The early work on the Imperial Standard Yard and its copies was carried out with the bars floating on mercury so that their weight should be evenly distributed and supported. Sir George Airy, then the Astronomer Royal, proved that the bars could be supported, without loss of accuracy, on eight rollers mounted on a compensating frame of the form shown in Fig. 3, which distributed the supporting reactions evenly. For a frame of 'n' rollers, Airy's formula for the spacing of the rollers was $L/\sqrt{(n^2-1)}$, where L is the length of the bar. It was subsequently found that the number of rollers could be reduced to two, when their separation becomes ·577L ; their positions are usually marked on the bar and are known as the Airy points. When supported at these points, the surface towards the ends of the bar lies in a horizontal plane and, although the bar inevitably sags in the middle to a certain extent, the error in length is negligible.

One or more of these point gauges are kept in the gauge stores to suit every standard diameter being machined. This results in very large stocks being housed, and the gauges have to be carefully

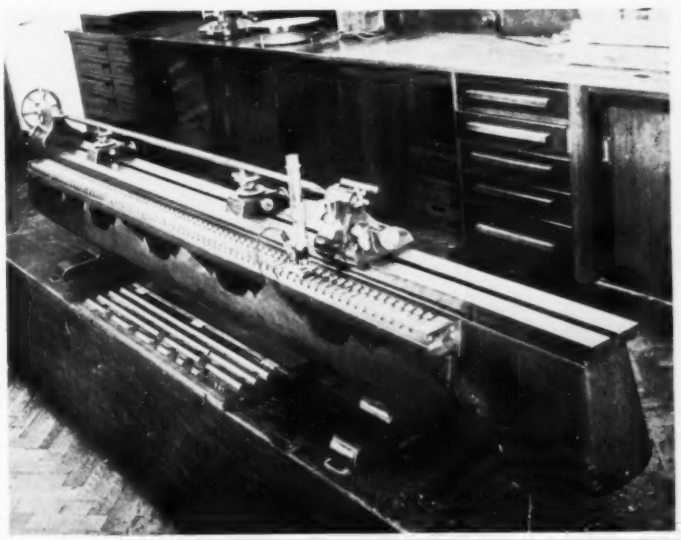


Fig. 5.—Newall length measuring machine

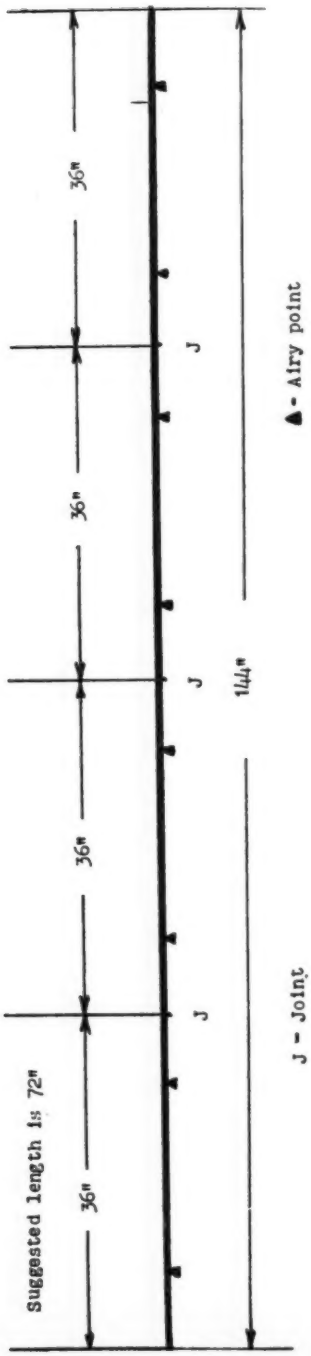


Fig. 6.—Series of length bars, end to end, each supported at its Airy points

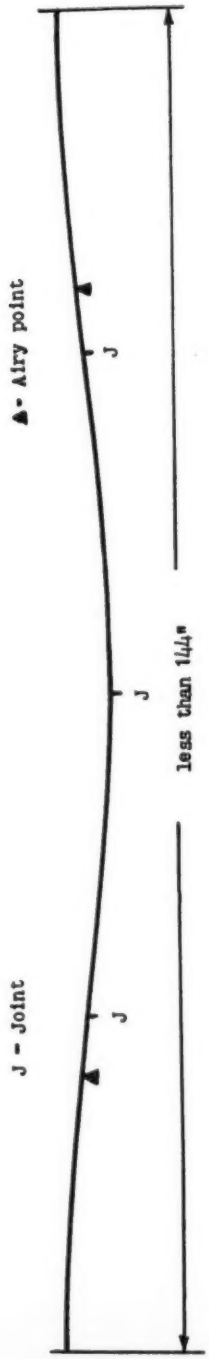


Fig. 7.—Composite bar supported at its Airy points

stored and checked each time they are returned to the stores. Stick micrometers up to about 6 feet (1,830 mm) in length, such as those made by the Pitter Gauge Company, are used for non-standard diameters.

Apart from these square-section point gauges suitable for measuring one diameter only, other gauges called telescopic point gauges are in constant use on diameters up to 20 feet (6,100 mm). Fig. 4 shows a few examples of this type of gauge.

The next consideration is the calibration of these shop point gauges, and the control of temperature is an important aspect. The obvious solution is to house length measuring machines in the standards room, but in order to stabilise the gauges during calibration, a loss of 48 hours at least would be required for each gauge, and this is impracticable. Therefore, these measuring machines are housed in the tool stores which have an overall temperature similar to the machine shops. Care must be taken that the length measuring machines are positioned away from draughts or shop heaters. Fig. 5 shows the type of machine used for calibrating the one-piece point gauges.

Long Telescopic Point Gauges

The method of calibrating these gauges, although the same as for the shorter one-piece point gauges, creates a problem due to the sag in both the gauge and the primary standard. The primary standard is of necessity made up of composite lengths—which it is suggested should each be 6 feet (1,830 mm) to reduce the number of joints—and it is advisable to support each length at its Airy points to remove any stress from the joint. The primary standard is built up with bars to suit the length of the point gauge to be calibrated.

A series of such bars placed end to end with each supported at its Airy points would be dimensionally correct from end to end, as illustrated in Fig. 6.

The composite bar supported at its Airy points, as shown in Fig. 7, sags and is less than the combined length of its components. Similarly, when the long point gauge is supported at its Airy points it also will sag.

The amount of this shortening is dependent on

(i) length of bar ;
(ii) material of bar ;
(iii) shape of cross section of bar.

Fig. 8 shows graphically this decrease or shortening for four different typical bars *all supported at the Airy points.*

It will be noted from the graph that the extent of shortening varies considerably with the type of gauge used, the most suitable type being one made from $\frac{3}{4}''$ (19·05 mm) standard-weight steel tube (Curve ' C ' on graph), because it is comparatively light and for the majority of applications is not subject to differential temperature changes, as steel is the most common material for heavy engineering components.

For example, a 20 feet (6,100 mm) long gauge made up from $\frac{3}{4}''$ (19·05 mm) pipe tube will only shorten in the order of ·0004″ (·010 mm) when correctly supported on its Airy points. This amount of shortening can be considered negligible in comparison with the overall length of the gauge.

The problems involved when calibrating these long point gauges against the standard composite workshop length bars are :—

(1) Design of length measuring machine.
(2) Effect of temperature.
(3) Location of length measuring machine in the machine shops.

(1) It is not considered, from experience, that an elaborate arrangement is required for comparing the lengths against standards. Fig. 9 shows a type of length measuring machine used for this purpose.

The beam is fixed at one end and supported along its length on balls to prevent distortion of the beam with temperature changes. The two blocks are adjustable and the button in the centre of each block is lapped flat and arranged with its face at right angles to the centre-line of the gauges to be measured. The standards and length bars are each fitted with spherical ends to give point contact to the button. When making calibrations, the standard composite length bars and shop point gauges are in turn set up on their Airy points, one end in contact with the button, the other end block being arranged with a ·01″ (·254 mm) gap so that a ' go ' and ' not go ' feeler gauge can be inserted ; by this

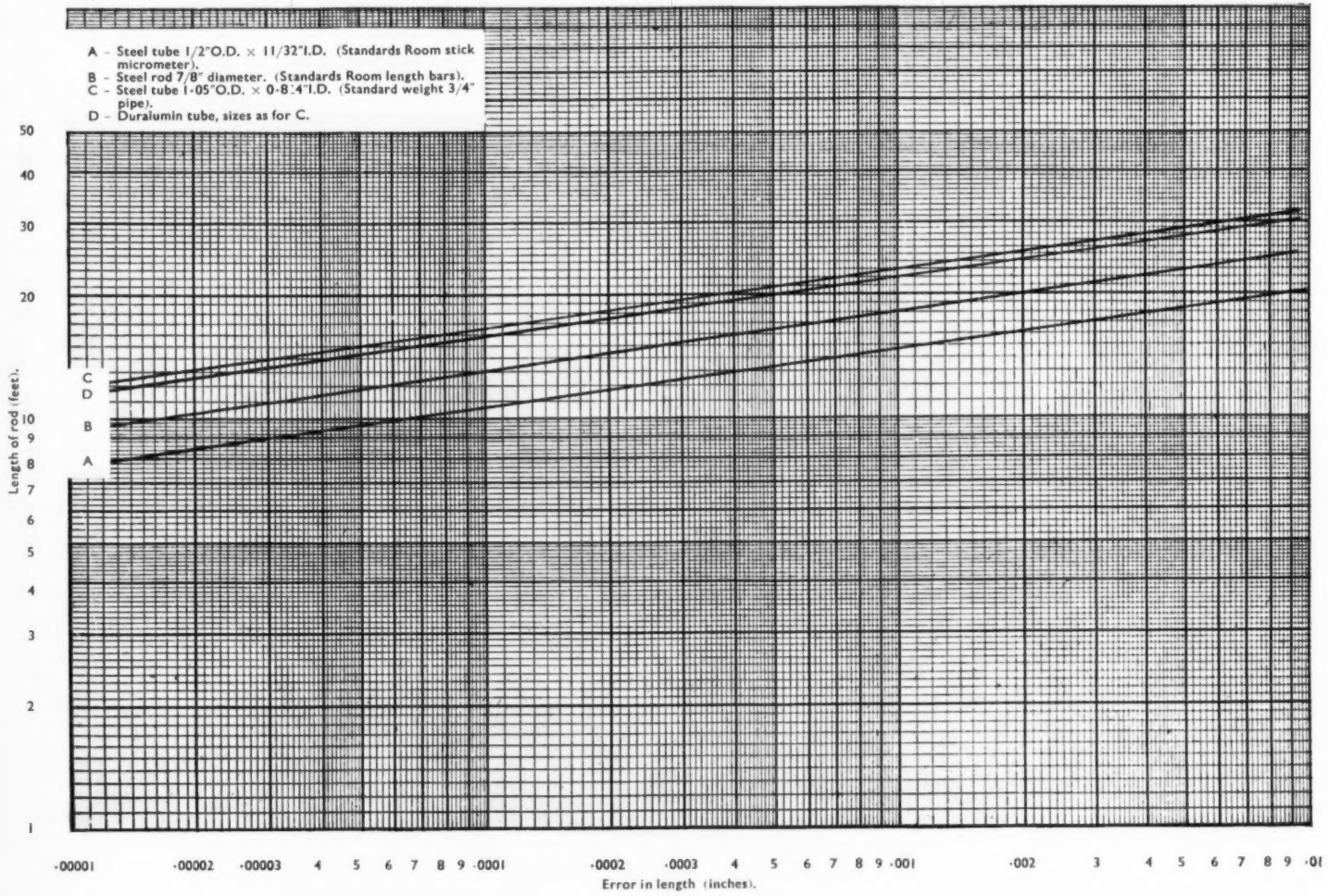


Fig. 8.—Graph showing decrease or shortening for four bars of various cross sections supported at their Airy points

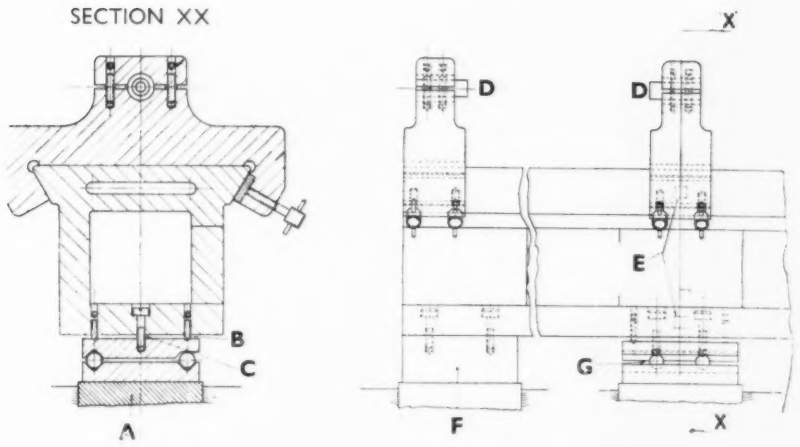


Fig. 9.—A 30-feet measuring bench

A. BASE HELD IN BRICKWORK PLATFORMS.
B. LEVELLING & LOCATING SCREWS.
C. SECURING SCREWS.
D. MEASURING HEADS TO ACCOMMODATE BUTTONS OR MICROMETERS.
E. KEYWAYS TO LINE UP BED SECTIONS.
F. FIXED END.
G. DOUBLE TRACK BEARINGS.

means the workshop point gauges can be adjusted to an accuracy of ·001″ (·025 mm) which is all that is required when dealing with diameters of 6 feet (1830 mm) and over.

(2) At this stage it will be useful to show the effect of temperature changes on gauges for measuring large diameters. Fig. 10 shows graphically the extent of the expansion and contraction of steel bars, for which the coefficient of linear expansion is taken as 0·00000636/° F (0·00001145/° C).

This graph gives the expansion or contraction for steel bars up to 30 feet (9150 mm) long for temperature differences of 2, 4, 6 . . . 18, 20° F (1·11, 2·22, 3·33 . . . . 10, 11·11° C). It will be seen that quite appreciable changes in length do take place with very few degrees of alteration in temperature.

Whilst this is not easy to control, it is simplified if the point gauges are made from steel with the same coefficient of linear expansion as the component being measured, and each point gauge is laid on the machine table for some hours before use. Further reference will be made to the effect of temperature later in the article.

(3) As already mentioned, a length measuring machine is for all practical purposes best housed in a room within the machine shops.

The foregoing remarks deal with the calibration of shop point gauges against standards, and as in the case of reference and workshop standards the importance of ensuring that such calibrations are correct cannot be too greatly stressed because the results are the only basis for future measurements in the machine shops. The practical error in determination of point gauges need not be greater than ·002″ (·050 mm) even on lengths of 20 feet (6100 mm), with corresponding reduction in error as the length decreases. Such errors can be considered negligible for all practical purposes.

Measurement of Internal and External Diameters of Machined Components

Following the calibration of the gauges, the problem of transferring the measurement from the point gauge to the machined component must now be considered. This is a highly skilled operation.

Dealing first with inside diameters, an exercise was carried out using skilled boring mill operators and experienced heavy machine shop inspectors from two separate machine shops to measure a diameter of approximately 70½″ (1790 mm). This operation was carried out in the standards room because it was felt that any unknown variations such as temperature should not enter into the exercise, as its purpose was to find the error of

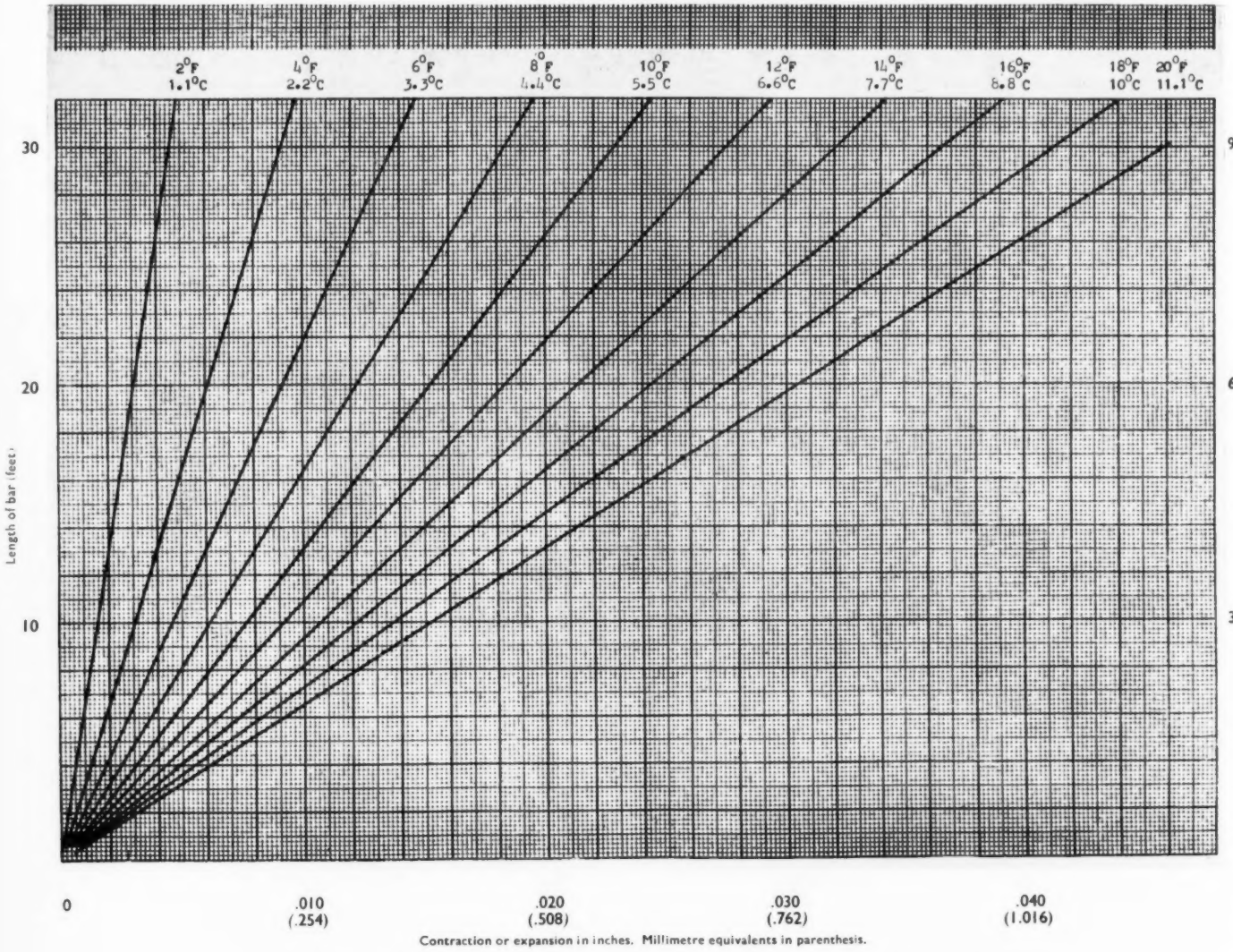


Fig. 10.—Expansion or contraction of steel bars for various temperature changes

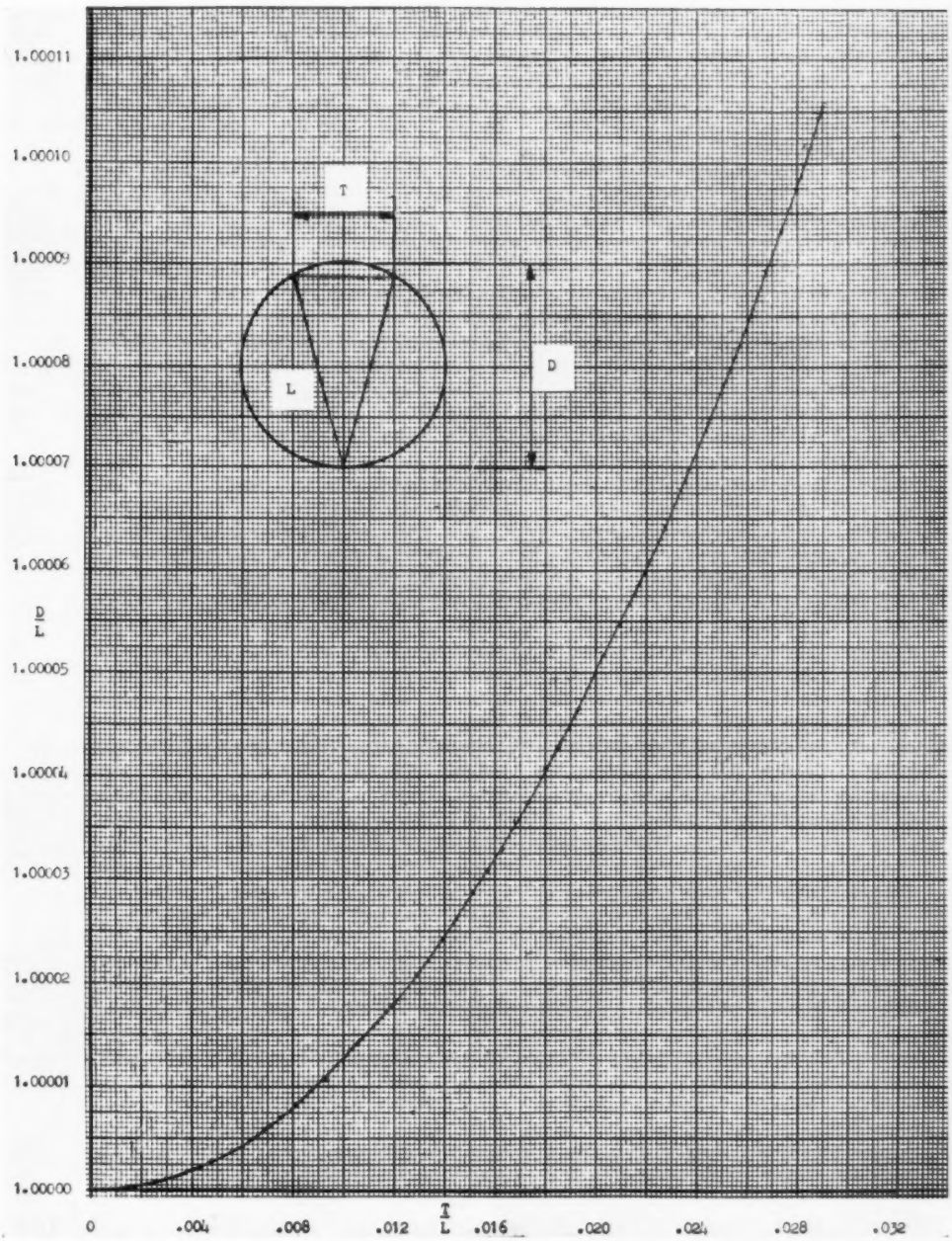


Fig. 11.—Relationship between end travel of point gauge T, its length L and bore D

determination between the operatives. The results obtained are tabulated below.

Operative	Shop Point Gauge	Stick Micrometer with Clock Indicator	Stick Micrometer only
Inspector	70·3282″ 1786·336 mm	70·3255″ 1786·268 mm	70·3294″ 1786·369 mm
Inspector	70·328″ 1786·331 mm	70·3255″ 1786·268 mm	70·3295″ 1786·371 mm
Inspector	70·3283″ 1786·339 mm	70·3255″ 1786·268 mm	70·3296″ 1786·374 mm
Inspector	70·3283″ 1786·339 mm	70·3255″ 1786·268 mm	70·3297″ 1786·376 mm
Boring mill operator	70·3283″ 1786·339 mm	70·3255″ 1786·268 mm	70·3293″ 1786·366 mm
Boring mill operator	70·3282″ 1786·336 mm	70·3255″ 1786·268 mm	70·3293″ 1786·366 mm

The maximum difference using the same type of gauge is only ·0004″ (·010 mm), but it will be noted that there is approximately ·004″ (·102 mm) difference between the three columns of figures. The chosen ring was oval and each type of gauge was used specifically across different sections of the ring.

Neither inspector nor operator knew what his colleagues had recorded. All these men were highly skilled, but even so, it is difficult to credit such a standard of accuracy.

This operation was repeated in another works of the Company by inspectors only, on a diameter of 43·782″ (1112·063 mm) using a Pitter stick micrometer, and in this case was carried out in the open shop. The readings are given below.

	Inches	Millimetres		Inches	Millimetres
1.	43·7822	1112·068	9.	43·7820	1112·063
2.	43·7822	1112·068	10.	43·7820	1112·063
3.	43·7822	1112·068	11.	43·7822	1112·068
4.	43·7825	1112·075	12.	43·7825	1112·075
5.	43·7822	1112·068	13.	43·7835	1112·101
6.	43·7820	1112·063	14.	43·7825	1112·075
7.	43·7822	1112·068	15.	43·7825	1112·075
8.	43·7822	1112·068	16.	43·7827	1112·081

The maximum difference between the readings is ·0015 (·038 mm), which is a more practicable result.

It will be appreciated that the same measuring stick was used in each case, which might have an error against the standards, and it is possible that some figure would have to be added or subtracted to obtain the correct size.

End-wobble of Point Gauge

After machining, there must be some ovality, taper or other deviation from true form, and an allowance should be made to make sure that the diameter is large enough everywhere to pass over its mating part. This is usually done by allowing the point gauge some transverse movement. During the foregoing checks this point was discussed and it was decided that $\frac{1}{8}$″ per foot (1 in 96 mm) travel was a satisfactory figure. In addition to allowing for inaccuracies in machining, the problem of ' feel ' would be largely overcome because it is easy for an operator or inspector to see a difference of $\frac{1}{4}$″ (6·35 mm) in travel. The graph in Fig. 11 shows how the difference in gauge length L from bore size D can be obtained for any amount of end travel T of the point gauge. It can also be seen by a simple calculation how little an extra $\frac{1}{4}$″ (6·35 mm) travel of the point affects the measurement of internal diameters ; hence we have in effect a very high magnification of reading.

Support of Point Gauges 6 Feet and Under

One man can quite easily operate the point gauge within a bore. He can support it with one hand at each of its Airy points, and gauges in this range are robust enough not to sag sufficiently to cause shortening by any measurable amount if not supported exactly at the Airy point position.

Supported Point Gauges Over 6 Feet

With these longer gauges it becomes necessary to use two men and, further, the Airy point positions on gauges 10 feet (3050 mm) and over are too far away from the ends for even two men to support them comfortably. In such cases it is advisable to build from the boring mill table two supports which can be raised or lowered in the bore, at the same time being kept as near as possible in the same horizontal plane ; a small amount of out-of-level does not matter in these large diameter bores.

In order to determine the effect of the Airy point supports being out of level, the graph shown in Fig. 12 indicates the errors that would be introduced if the supports were incorrectly adjusted, e.g. the 10 feet (3050 mm) point gauge with its Airy points differing in height by ·2″ (5·08 mm) would give an error in reading of ·0005″ (·013 mm), and it is easy to control the level much closer than ·2″.

Fig. 13 shows a ring having the bore measured with a stick micrometer held in position by two operators, and Fig. 14 shows it being measured

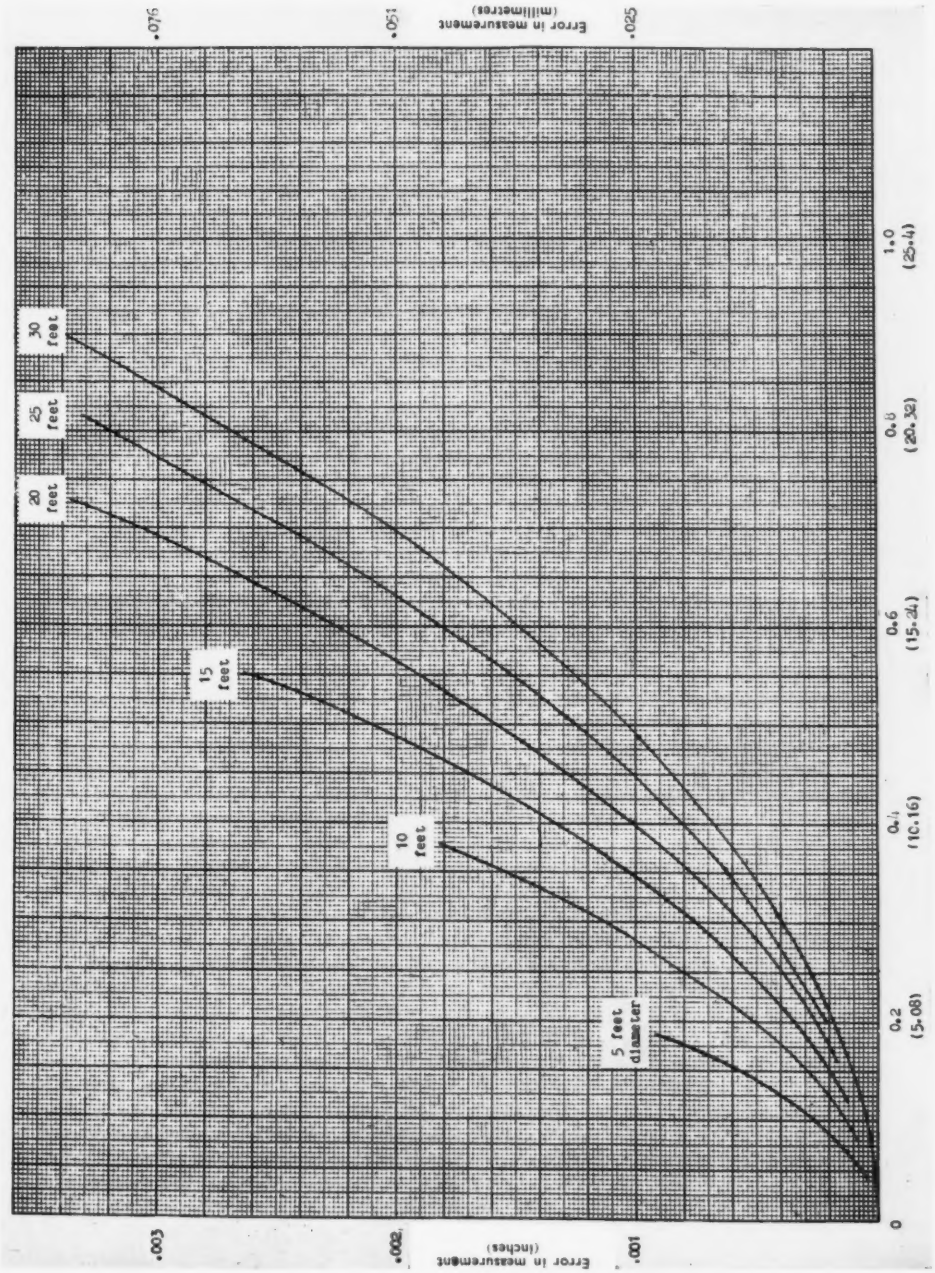


Fig. 12.—Error in determination of various diameters when the Airy points differ in height

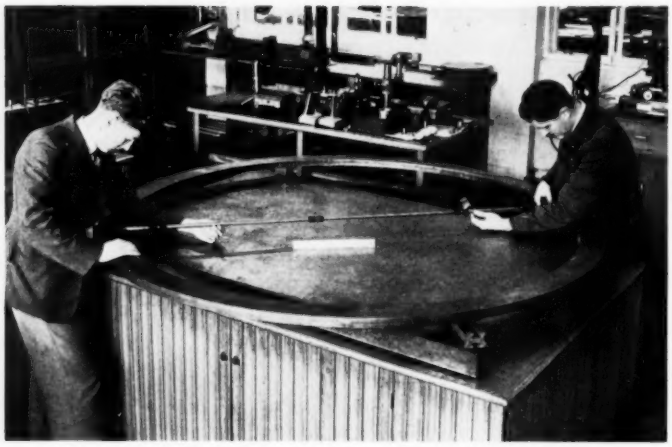


Fig. 13.—Measurement of a bore with a stick micrometer supported by hand

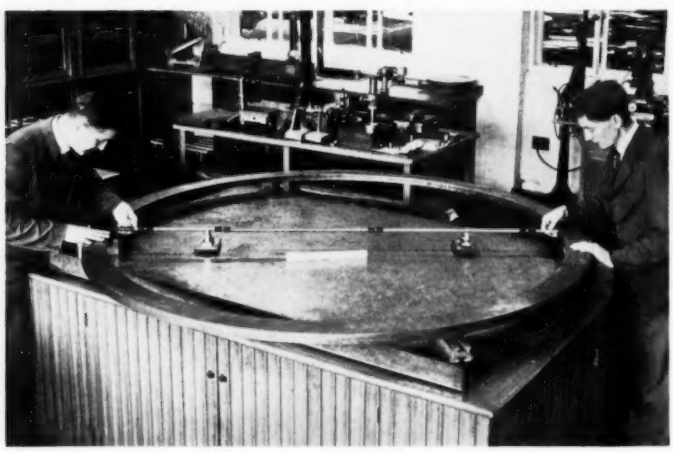


Fig. 14.—Measurement of a bore with a stick micrometer supported on floating platforms at its Airy points

when the same stick micrometer is rested on its Airy points in V blocks each mounted on a floating platform. These platforms consist of a plate mounted on captive ball bearings on a fixed second plate, and by this means the upper plate is free to move in any direction in the horizontal plane.

Supporting the point gauge in the manner shown in Fig. 17 allows the two operators freedom to work at each extremity of the gauge and reduces the danger of distortion ; one man merely holds the spherical end of the gauge lightly against the bore and the other man determines the lateral travel. Experiments have shown that accurate repeated readings can be obtained with this arrangement.

Again the design of the spherical end of the point gauge is important, because it is much easier to determine the extent of lateral movement with the end of the gauge approaching a point. Also the point of contact should always be on the axis of the gauge.

Fig. 15 is a typical example of large diameters which are being measured every day. In this illustration the bore of a large frame is being measured and the size established by two men supporting the point gauge by hand, which can hardly be described as comfortable. Readings 'A' below show the diameter of the bore taken in this manner at 12 o'clock and 3 o'clock and repeated in three places down the bore at each cross section. Fig. 16 shows exactly the same diameter being measured using the same point gauge but supported at its Airy points on floating platforms. Readings **'B'** show the results obtained with this setting.

Readings A	12 o'clock		3 o'clock	
	Inches	Millimetres	Inches	Millimetres
Top.. ..	146·100	3710·940	146·101	3710·965
Centre ..	146·1035	3711·029	146·1015	3710·978
Bottom ..	146·1095	3711·181	146·100	3710·940
Readings B				
Top.. ..	146·102	3710·991	146·104	3711·042
Centre ..	146·104	3711·042	146·103	3711·016
Bottom ..	146·106	3711·092	146·102	3710·991

It will be seen from these readings that there is some difference with the two methods both in size and shape, and it is the author's intention to continue investigating the two methods by comparing measurements on as many large frames as possible.

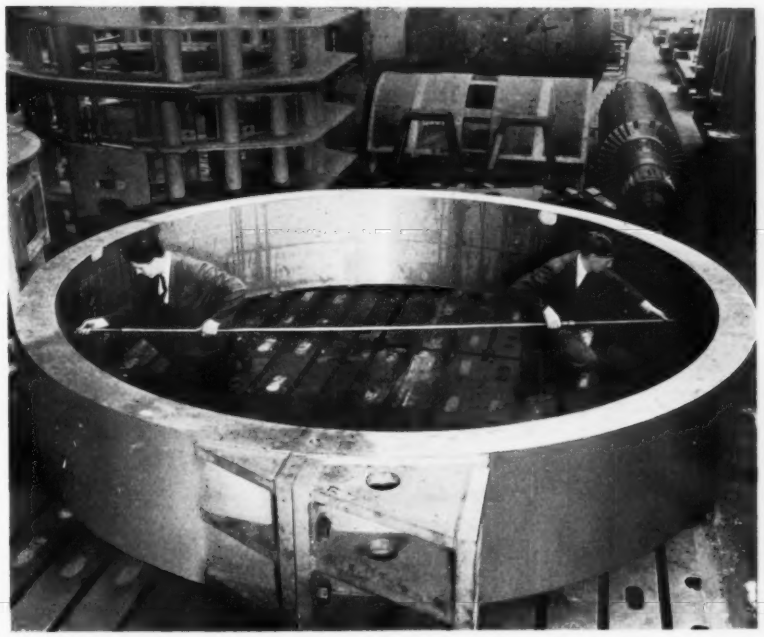


Fig. 15.—Measurement of 12-feet bore with point gauge supported manually

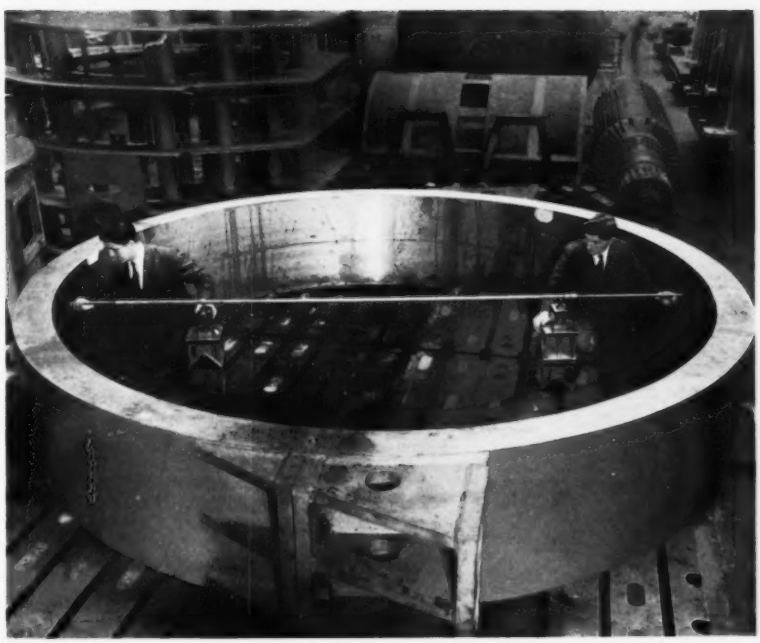


Fig. 16.—Measurement of 12-feet bore with point gauge supported on floating platforms at its Airy points

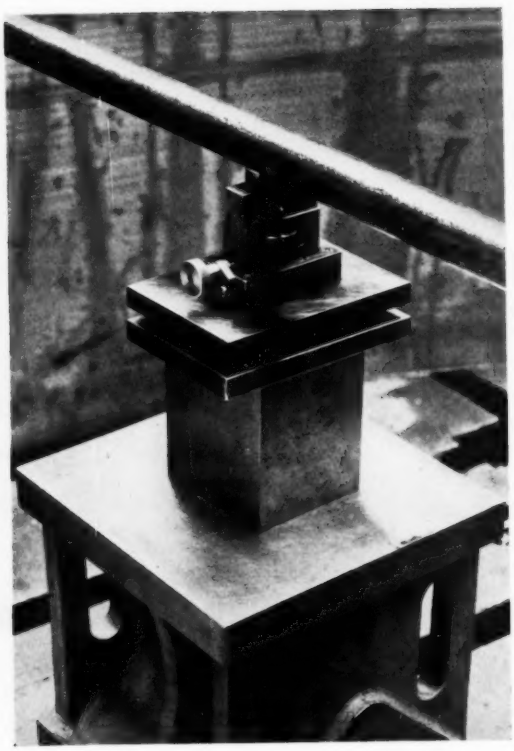


Fig. 17.—Close view of floating platform shown in use in Fig. 16

Temperature Control during Machining and Measuring Large Components

Temperature control of large components is of great concern to all connected with their measurement, and it may be that consideration will have to be given to the temperature control of the whole machine shop, but this largely depends on whether future requirements demand interchangeable parts to be made in many engineering shops in different parts of the country.

Large electrical frames are machined on boring mills, and it will be appreciated that a boring mill operator has difficulty in determining the final size because the metal is warm due to the heat generated by the tool during the cutting operation, and in all cases the bore is slightly larger than it will normally be when restored to a temperature around 68° F (20° C).

These boring mill operators, however, are very experienced men and during years of manipulation of tools of this type they have reached a position where they can make a very intelligent guess of the amount of metal to take off to bring the diameter between the drawing tolerances at 68° F. More accurate results can, of course, be achieved by taking the component off the boring mill, or even leaving it standing on the boring mill table for two days, before taking the final cut, but it will be appreciated that this would be quite impracticable in an engineering shop because of the loss of this time.

Therefore, before machining the mating part, the usual practice is again to measure the first part, take note of the temperature, and give a size to which the mating part is to be machined, the all important consideration being the relative fit between the two parts.

Measurement of Outside Diameters

The measurement of large outside diameters provides a much more difficult problem because the maximum diameter cannot be measured with semi-circular frame gauges such as large micrometers without the gauge flexing, which results in inconsistent readings.

Fig. 18 shows the construction of the general type of gauge in use, which consists of a wooden beam about 2″ × ½″ (50 × 12 mm) in section with a short caliper leg bolted at each end. Readings are taken of the diameter at each end of the piece being machined, extending along the length as far as the caliper legs will allow. The centre portion which is outside the range of the caliper legs may be checked by laying a straight-edge on the surfaces parallel to the axis and resting across the checked surfaces.

A good deal of skill is required to handle these gauges if consistent results are to be obtained. They should be handled with the same touch as that of a professional billiards player using his cue. There must be no strain ; only the finest touch should be felt when passing the caliper legs over the major diameter. When the operators have established a ' feel ' which satisfies them, the length between caliper legs is obtained from a similar check with the shop point gauge, as shown in Fig. 19. An exercise was carried out by a number of operators and inspectors checking a marked

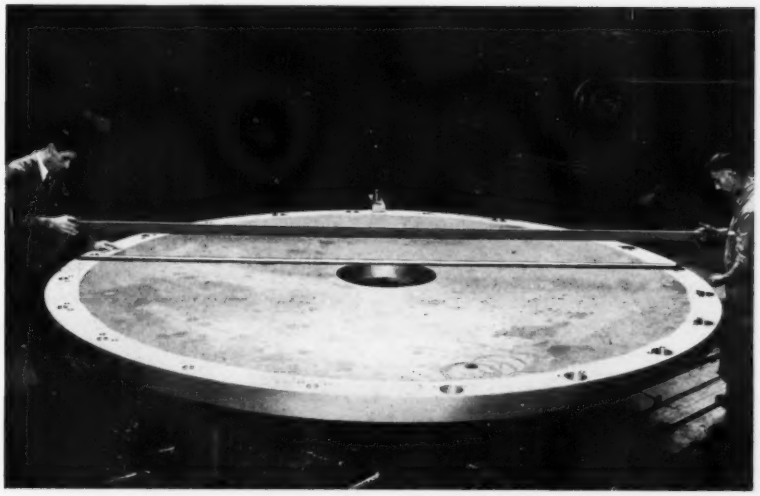


Fig. 18.
Use of a trammel to measure a 12-*feet diameter plate for a salient pole alternator rotor*

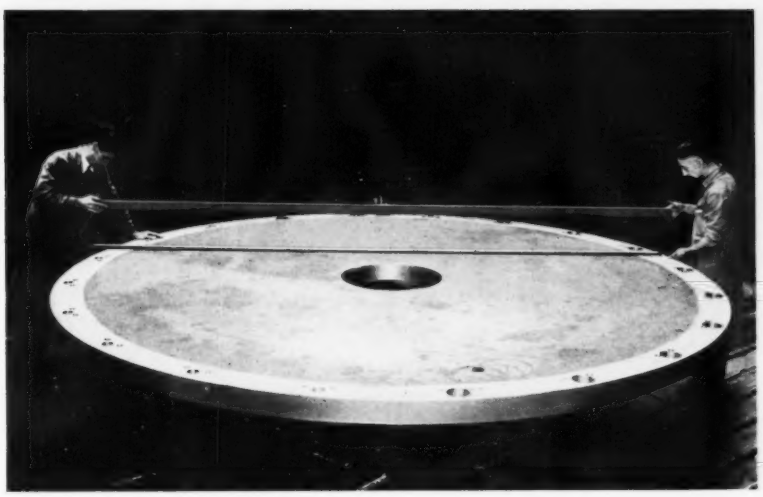


Fig. 19.
Applying the trammel size, established as shown in Fig. 18, *to a point gauge calibrated to length*

diameter on a ring of 77·5″ (1968·5 mm) diameter in the standards room, as shown in Fig. 20.

Results of readings are given below.

	Inches	Millimetres		Inches	Millimetres
1.	77·5250	1969·135	7.	77·5240	1969·110
2.	77·5240	1969·110	8.	77·5240	1969·110
3.	77·5235	1969·097	9.	77·5250	1969·135
4.	77·5240	1969·110	10.	77·5240	1969·110
5.	77·5235	1969·097	11.	77·5245	1969·122
6.	77·5240	1969·110	12.	77·5240	1969·110

The readings are so astonishingly close that the author questions their validity and proposes to follow up this exercise by taking readings of many more examples in the machine shops, as also intended for the inside diameter sizes. He also feels that considerable development can be made with the use of wood for outside diameter measurement.

A further exercise was carried out using a large external micrometer having a standard aluminium frame, and the figures obtained when checking a 34½″ (876·3 mm) diameter ring in the standards room are tabulated below ; it will be noted that they only differed around ·001″ (·025 mm).

	Inches	Millimetres		Inches	Millimetres
1.	34·5715	878·116	6.	34·5717	878·121
2.	34·5718	878·124	7.	34·5715	878·116
3.	34·5715	878·116	8.	34·5709	878·101
4.	34·5717	878·121	9.	34·5718	878·124
5.	34·5710	878·103	10.	34·5717	878·121

Fig. 21 shows this diameter being checked with an external micrometer and it will at once be noticed that the operator does not appear to be in a comfortable position, in fact it may well be described as awkward, which makes the accuracy of the above results all the more surprising.

A further set of readings was taken on a machined diameter of 46·283″ (1175·588 mm). Seventeen inspectors took the readings using a large micrometer in the open shop and the results are shown below.

	Inches	Millimetres		Inches	Millimetres
1.	46·2835	1175·601	9.	46·2830	1175·588
2.	46·2835	1175·601	10.	46·2815	1175·550
3.	46·2830	1175·588	11.	46·2827	1175·581
4.	46·2840	1175·614	12.	46·2832	1175·593
5.	46·2822	1175·568	13.	46·2835	1175·601
6.	46·2840	1175·614	14.	46·2837	1175·606
7.	46·2840	1175·614	15.	46·2810	1175·537
8.	46·2820	1175·563	16.	46·2802	1175·517
			17.	46·2822	1175·568

The maximum difference is ·0038″ (·097 mm) which is nearer to what might be expected.

Experiments have been carried out using as an alternative to this form of measurement a Chesterman tape rule graduated in thousandths of an inch on a Vernier scale. Using the 77·5″ (1968·5 mm) diameter ring, a comparison was made in the standards room by applying the tape around the diameter, a spring balance being used at the end of the tape to control the tension at 5 lb. Ten different operators took readings and it will be noted from the figures given hereunder that the difference between the maximum and minimum reading is ·001″ (·025 mm) which rather indicates that this type of measurement is a little more accurate.

Fig. 21.—A 34½″ diameter being measured with an external micrometer

Fig. 20.—Checking the diameter of a 77½″ outside diameter ring with a trammel

	Inches	Millimetres
1.	77·5275	1969·198
2.	77·5250	1969·135
3.	77·5265	1969·173

	Inches	Millimetres
4.	77·5270	1969·186
5.	77·5270	1969·186
6.	77·5265	1969·173
7.	77·5270	1969·186
8.	77·5270	1969·186
9.	77·5275	1969·198
10.	77·5270	1969·186

Unfortunately, however, these tapes only measure the length of perimeter; they give no indication whatever whether the job is round, and where it is practicable to use this type of measuring tape it should always be done in conjunction with a point gauge and wooden outside caliper gauge in order to obtain the extent of ovality.

The use of these tapes is very limited in engineering shops because a great number of the large constructions have interrupted diameters such as shown in Fig. 22.

Specially constructed gauges as seen in Fig. 23 can be made up to take readings more accurately, but generally this type of gauge is useful only on repetition work where it can be accurately set up to measure one given diameter. Equipment of this kind is in constant use where 'numbers off' are required.

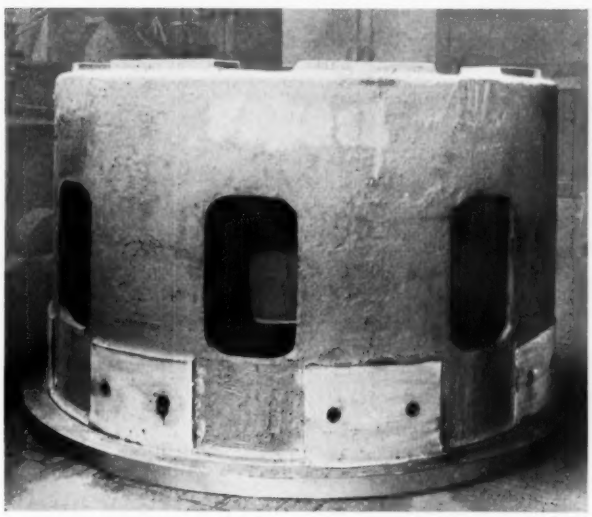


Fig. 22.—Upper bracket centre for a vertical alternator, showing interrupted diameter at the bottom. The casting is about 62″ high

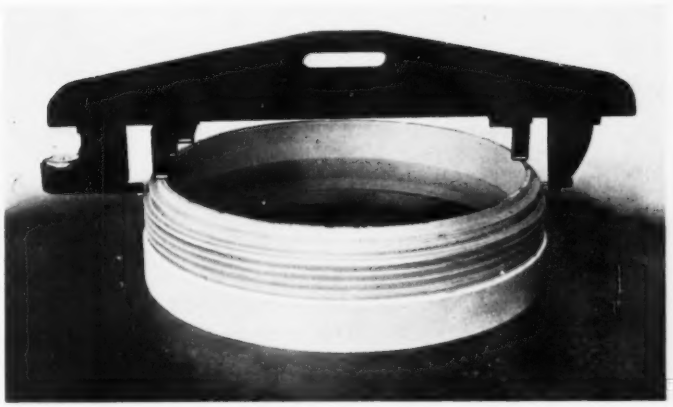


Fig. 23.—Special gauge for measurement of outside diameters

General

It is difficult to assess the full value of the readings recorded in this article because, due to the special nature and conditions of the exercises described, they do not exactly correspond in accuracy to the sort of measurements taken by the operators during their ordinary daily duties.

It is quite a different exercise asking an inspector to check any machined component against a drawing dimension. Usually he is right, but it has been found on occasion that he is wrong. Therefore, it is important that the design of the equipment should be such that the operation of checking is simple. The checks that have been taken show that whilst the machining of an internal diameter is more difficult than an external diameter, the measurement of an internal diameter is the easier problem. The introduction of a floating platform assembly for supporting the point gauge when taking internal checks will do much to minimise incorrect measurements.

The question of interchangeability becomes very difficult on sizes above 3 feet (915 mm). It can be done but the tolerance allowances on the mating parts would have to be in the order of ± ·003″ (·076 mm) for 3 feet (915 mm) diameter, and ± ·006″ (·152 mm) for, say, 6 feet (1830 mm) dia-

meter ; in addition, an allowance would have to be made by adjusting the dimensions so that spigot fits would always have a clearance. Engineering shops therefore resort to the principle of machining one part, measuring the diameter carefully and giving a size for the mating part. Usually a further precaution is taken to use the same skilled inspector for each component. The inspector must watch the vexed question of temperature difference when making his checks.

Designers in engineering works try to appreciate these problems and consequently make an allowance for final adjustments in concentricity or alignment.

There will, however, be a large number of cases nationally where parts have to be manufactured in different workshops in the country, and they must fit together. The author suggests that the simplest way to deal with such cases is to make the sub-assembly in one workshop and resolve the problem down to an accurate machined dimension which has to fit another sub-assembly from a different works. Arrangements can then be made to machine these accurate parts finally in a temperature controlled room, and check them with suitable precision measuring equipment.

An effort has been made within the limited scope of this article to give an insight into the difficulties that confront the shop-floor operators and inspectors when faced with the problem of machining and measuring parts for heavy electrical or mechanical engineering equipment. There is no doubt a great deal of scope for investigation on this subject, and considerable improvement could be achieved by collaboration between the gauge makers and large engineering firms in regard to the design of equipment for such measurement.

It should be appreciated that most large engineering works make only a few of each type of product, and therefore the measuring equipment must be flexible, as for example, the wooden beam and caliper legs.

Summary

From the contents of this article it will be noted that the three basic principles to follow are :—

(1) Correct calibration of the basic standards.

(2) Accuracy of measurement in machine shops.

(3) Control of temperature.

(1) and (2) can be taken care of satisfactorily with ordinary precautions, but (3) is much more difficult and will therefore have to be given a great deal more consideration before a solution is reached.

The Napier Surface Heater De-icing

A PRACTICAL SOLUTION TO THE AIRCRAFT ICING PROBLEM

THE AIRCRAFT ICING hazard has been a recognised problem since the early days of flying and has recently been accentuated by the general tendency towards all-weather operation, higher operating altitudes, higher average speeds and the greater susceptibility of the modern aircraft to ice accretion. This increase in all-weather flying tends to lengthen the total time during which an aircraft may be subjected to icing conditions, and the higher flying speeds cause a corresponding increase in the rate of accretion. These factors, coupled with the intolerance of turbine engines to ice, can lead to icing beyond the control of protection equipment hitherto available, thus dislocating operational services and imperilling the safety of aircraft.

In the course of flight testing propeller-turbine engines, D. Napier and Son Limited encountered these difficulties and initiated an extensive programme of research and development to overcome them. This has resulted in the Napier electrical system of de-icing incorporating a sprayed heater mat of unique construction.

The Viking test aircraft upon which the initial development of the Napier heater mat was carried out is shown in Fig. 1, which illustrates the test heater mats mounted on the fin leading edge and the water spray mast for producing artificial icing conditions.

An Outline of the Aircraft Icing Problem

Ice will form on an aircraft when flying through clouds at temperatures below freezing because the droplets which constitute the cloud are capable of existence as supercooled water. When these droplets strike the surface of the aircraft the transition from liquid water to ice is initiated. The rate of freezing and the characteristics of the resultant ice formation are dependent on the heat transfer conditions in the region of the impact, and

Fig. 1.—*The Viking test aircraft equipped with a water spray mast and a test heater mat on the fin leading edge*

the meteorological conditions encountered.

Ice formation on an aircraft during its passage through supercooled cloud is known as 'impact icing', and weather conditions play an important part in determining the severity of ice accretion. Supercooled water droplets can exist in the atmosphere at temperatures down to −40° C and at altitudes from sea level to 40,000 feet, but fortunately the lower temperatures are normally associated with smaller concentrations of liquid water. Icing clouds, however, can extend over very large areas and to a considerable depth ; consequently any protective system installed must be capable of continuous operation for long periods.

The effects of ice can be serious in many respects. Accretion on the aerofoil surfaces changes their contour, causing a reduction of lift coefficient and an increase of drag and weight. Engine and other auxiliary intakes may become partially or completely blocked, and pieces of ice formed in an engine intake may break away and enter the engine, resulting in serious damage, particularly to axial flow compressors. Windscreens may become completely obstructed, and radio masts, aerials and flying instruments may also be affected.

Some components of an aircraft are more susceptible to ice formation than others, whilst certain components are particularly vital to the safety of the aircraft. The engines for example, particularly turbine engines with axial flow compressors, fall within both these categories and hence must always be a primary consideration for protection as an insurance against chance encounters of icing conditions severe enough to be disastrous. The aerofoil surfaces of an aircraft are less critical due to their larger scale causing a slower relative rate of accretion and a certain degree of tolerance to relatively large deposits, although great care must be exercised during landing with an ice formation on the flying surfaces. The extent to which any particular aircraft is protected, therefore, is largely a matter for the operator to decide, the degree of protection being dependent on the rôle of the aircraft and the economy of its operation.

In the past, mechanical and chemical systems of icing protection have been used, but as the majority had severe limitations, thermal methods have been evolved which overcome most of these limitations. The National Advisory Committee for Aeronautics in the United States, the National Aeronautical Establishment in Canada and the Royal Aircraft Establishment in this country have carried out the majority of the fundamental research in this field, thus enabling thermal systems to be designed and operated with a high degree of confidence in their effectiveness.

The Thermal Method of Ice Protection

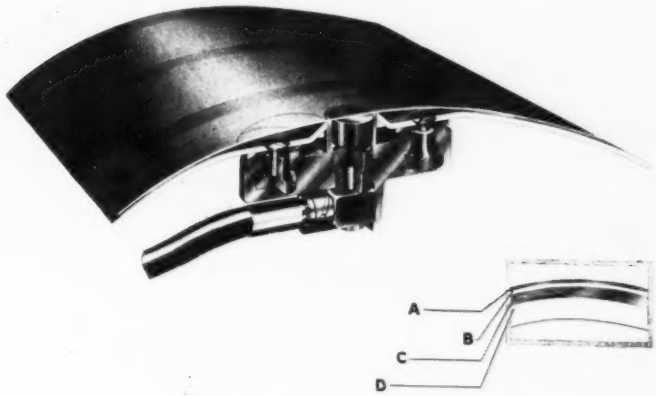
In the adoption of thermal methods of protection for aircraft, there are two principles of operation available. One termed 'anti-icing' is to prevent any ice forming at all, and this can be further sub-divided into wet anti-icing and dry anti-icing. In the wet anti-icing method the water caught is kept just liquid and will run back and tend to freeze behind the protected area ; it is thus limited in application. With dry anti-icing, all the water caught is completely evaporated.

In the second principle, termed 'de-icing', ice is allowed to form and is shed periodically before it can reach sufficient proportions to have a serious effect. This method can be adapted to any areas where the shedding of ice will not have a secondary detrimental effect, such as the main flying surfaces of the aircraft, the propellers, and in the case of helicopters, the rotor blades. A very great thermal economy is achieved since the heating is intermittent, the actual saving being dependent on the ratio of 'heat on' and 'heat off' time, both of which can be varied to suit any particular application. A typical ratio is 1 : 10, and because sufficient heat is applied to melt only the interface between the ice and the aircraft surface, the total power requirements can be as low as 1/20 of those required for anti-icing.

With the introduction of light-weight generating equipment with specific outputs of better than ½ kilowatt per lb weight, the use of electrical surface heaters is now well established for aircraft protection.

When the energy required for an application of thermal anti-icing or de-icing is provided by electrical means, perfect distribution of energy is possible with the ability to grade the heat to suit the requirements of any particular surface. As there is no complicated ducting or double skinning

to consider, the installation is extremely simple and can be installed on existing aircraft with few, if any, structural alterations. An electrical system is particularly suitable for propeller protection and for small auxiliary intakes, where protection by any other means would be practically impossible. The thermal efficiency of an electric heater mat is very high, since it can be applied directly to the outside of the surface to be protected and is therefore in direct contact with the offending ice. Exact control and regulation of heat is also possible, which together with the very small thermal lag makes electrical heating ideally suited for de-icing work. This small thermal lag, which is of the order of 5 seconds, is extremely important in de-icing to permit rapid heating and cooling and thus obtain clean shedding without run-back of water.



A Outer Insulation
B Sprayed Metal Element
C Base Insulation
D Base Material

Fig. 2.—The Napier heater mat construction and terminal assembly

By early consideration of the electrical system as a whole, in conjunction with the requirements of the icing protection system, load sharing arrangements can be devised making considerable weight saving possible. For example, the undercarriage can be operated electrically or in certain cases the alternator might well be used as an engine starter motor. In cases of long endurance in icing conditions, electrical protection shows an advantage even when the whole of the weight of the generating equipment is debited against it, for fuel consumption figures make the curve of total weight against endurance in ice rise quite steeply for a hot gas system, whereas the curve of an electrical installation is considerably flatter.

Technical Requirements of an Electric Heater Mat

The requirements of a heater mat for ice protection applications are :—

1. The thermal efficiency shall be high, *i.e.* the heat losses to the structure shall be a minimum and the heat transfer to the external surfaces shall be a maximum.

2. The resistance of the conductor element shall be capable of adjustment in the design stage to give any required power loading at any specified supply voltage over a wide variation of areas.

3. The physical, electrical and mechanical properties of the heater shall be substantially unaffected by temperatures within the range of -60° C to $+70^\circ$ C, and it shall be non-inflammable.

4. The external surface finish shall be as smooth as possible for aerodynamic reasons. It shall also have a high resistance to abrasion and rain erosion.

5. Its thickness shall be a minimum to avoid undesirable changes to the surface contours and the structure generally.

6. It should be capable of application to any shape of surface including those of compound curvature.

7. Good adhesion to metallic surfaces and to light alloys in particular is essential.

8. The insulation material shall have low water absorption properties and be impervious to aircraft fuels, oils and hydraulic fluids.

9. It should be possible to carry out local repairs on the heater, if necessary in the field.

10. Its weight per unit area shall be a minimum.

The above requirements for a heater mat show that it is advantageous to fabricate the heater by a spraying process directly on to the surface to be protected, particularly when compound curvatures and the difficulties of applying a prefabricated mat without entrapping air at the attachment face are

considered. This trapped air is liable to cause blistering and overheating when heat is generated by the elements, but a sprayed application of the base insulation ensures intimate contact with the surface. Similar considerations also show that the electrical element itself is advantageously applied by a spraying technique, which also satisfies the requirements by permitting adjustment of the resistance to any specified power loading during the design stage.

The ' Napier ' Construction of Heater Mat

Research and development on the foregoing lines at the Napier Flight Development Establishment has led to the heater mat construction illustrated in Fig. 2. The base insulation consists of a layer of a thermo-setting resin applied by a flame spray gun, over which is applied the metal electrical conductor. This is also applied by a flame spray gun to the resistance and pattern called for by the particular heat distribution required, and terminals are provided for connection into the electrical system. The outer insulation, of similar material to the base layer, is sprayed on and finally finished as a highly polished surface. The surface finish of the heater mat may be coloured if required without affecting the characteristics of the mat in any way.

The Napier heater mat may be applied to the surface of suitable shapes by mechanical means, which normally take the form of gun traversing gear linked by a mechanism to the work piece turn-table in such a way that the rate of pass of the gun across the work piece can be controlled. The heater element is sprayed in a similar way, close control being kept over all spraying conditions to ensure that the required overall resistance and degree of uniformity is obtained. In general, objects of single uniform curvature, of aerofoil section, and of spinner section can be treated by mechanical methods, providing the quantity required warrants the cost of mechanisation.

For more complex shapes, a combination of hand and mechanical methods is normally employed. In both cases the process may be broken down into six main operations :—

(*a*) Terminal fitting.
(*b*) Base coat application.
(*c*) Element application.
(*d*) Top coat application.
(*e*) Testing.
(*f*) Final finish.

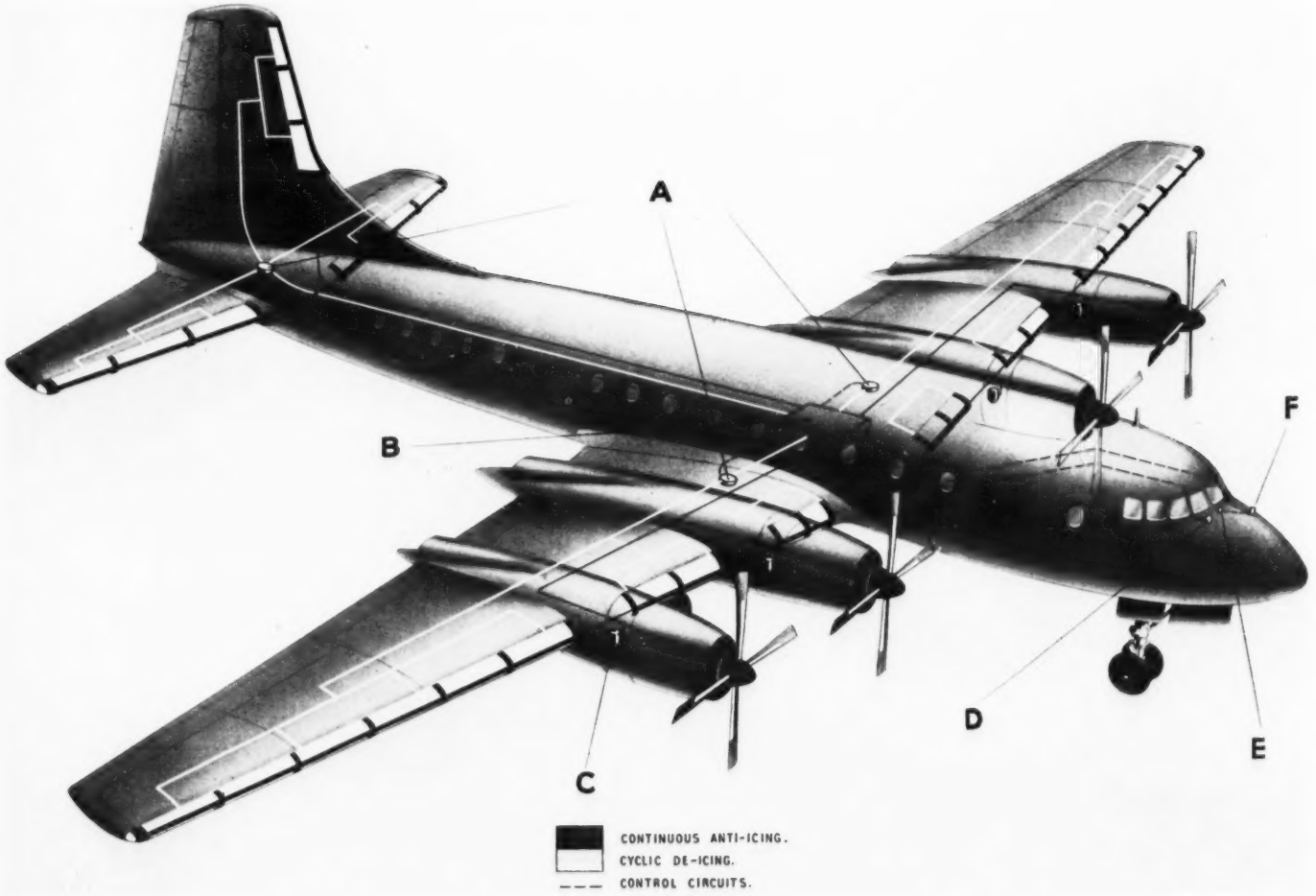
A range of standard terminals is available of the type illustrated in Fig. 2, designed to suit the system of wiring of the Society of British Aircraft Constructors.

A very light heater mat is achieved by this method of construction, its total thickness being ·040 to ·050 inch depending on the conductor thickness, and its corresponding specific weight being 0·27 to 0·32 lb/ft² allowing one terminal per square foot.

The heater mat has been extensively tested for general serviceability and shown to fulfil the requirements to a very high degree, having successfully undergone a Ministry of Supply approved type test consisting of the climatic tests of DTD. 1085B, and the vibration, electrical insulation and inflammability tests of British Standard G.100 for Class 1 airborne equipment. Other tests include powered endurance tests in which 1,000 hours of cycled operation at 10 watts/in² with water cooling, and 1,000 hours continuous and 1,000 hours cycled operation at a surface temperature of +70° C have been successfully achieved. With cold-air wind tunnel cooling, power loadings of 40 watts/in² have also been maintained without failure. The heater has proved to be unaffected by aircraft fuels, oils and hydraulic fluids, or by the new ester-based lubricants which are now being adopted on turbine engines. Flexural fatigue tests carried out with mats mounted on resonant bars have shown that the element imposes no stress limits on structures, and permits stressing of the parent metal up to the Wöhler fatigue limit. Type tests, consisting of full power tests, tests at +70° C and electrical insulation tests, are also carried out on at least one sample of any specific installation.

The maximum operating temperature for the heater mat is at present fixed at 100° C, and whereas this is more than adequate for use in icing conditions where cooling is severe, it is normally advisable to install some form of overheat protection to safeguard the heater in cases of less severe cooling, for example, on emergence from icing conditions into dry air. A thermal cut-out utilising a resistance sensing element has been developed for this purpose, which will also eliminate the human element during functional ground testing.

The resistance of the heater mat to sand abrasion



A Cyclic Switches
B Aircraft Power Supply Junction Panel
C Power Supply
D Cockpit Controls
E Pilot Warning Lamp
F Ice Detector for Automatic Operation

Fig. 3.—A typical application of electrical ice protection on a four-engined transport

has been proved by tests on York, Tempest and Canberra aircraft, including considerable service in tropical conditions, while the rain erosion properties of the standard insulation have also been established by 220 hours high speed tests in rain by a Venom aircraft. For high speed applications, use is made of a final outer layer of a modified resin (pigmented if required) which accelerated tests have shown to have superior erosion properties, and for yet higher speeds the application of a metal protective layer is adopted.

These mats have been proved by over 1,500 airborne hours during flight development and have flown 10 million miles in civil airline service.

Installation and Power Requirements

A typical all-electric protective system for a hypothetical four engined turbo-propeller transport is shown in Fig. 3. The areas particularly suited to the application of the Napier surface heater mats are the mainplane, tailplane and fin for de-icing, and the engine intake and propeller spinner for anti-icing. The propellers would be electrically de-iced and the windscreen anti-iced. Although the mainplanes shown are completely protected, it is not always necessary to de-ice the inner wing sections of a large aircraft. Table I shows the various loads required for this case, these being divided into continuous and cyclic loads. From a consideration of the design meteorological conditions, aircraft geometry and a selected safe allowable thickness of ice accretion, a cyclic ratio of ' heat on ' to ' heat off ' time of 1 : 10 has been selected for the cycled areas of the mainplanes, tail units and propellers. The mainplanes are thus conveniently divided into 20 areas with a continuously heated parting strip along the leading edge, and chordwise continuously heated dividing strips to ensure clean shedding of each cycled area. The heat requirement for these continuously heated zones is that for wet anti-icing, i.e. just sufficient to prevent bridging of the gaps by ice, and the requirement for the cycled areas is that sufficient to break adhesion and shed quickly and cleanly to prevent run-back. There is, of course, an optimum specific power loading for any given case, this being largely dependent on the conducted losses through the ice accretion and on the aircraft structure. The power required for protection of the engines is included and would probably be for dry anti-icing, as run-back cannot normally be tolerated in an intake.

The operation of the complete system for the initial switching on, variations of ' heat on ' and ' heat off ' times of the cyclic areas, and for temperature control of the continuously heated areas, would be fully automatic by means of an icing indicator and suitable temperature sensing elements. Pilot warning and override controls would of course be provided.

TABLE I

SUMMARY OF LOADS FOR ELECTRO-THERMAL PROTECTION FOR A TYPICAL FOUR ENGINED TURBO-PROPELLER TRANSPORT

Weight—120,000 lb. Span—130 ft. Cruising speed—350 m.p.h. at 20,000 ft.

Component	Power Input		Remarks
	Continuous	Cyclic	
Wings ..	26·5 kW	10 × 32·8 kW	Assuming no protection inboard of inboard nacelles.
Tailplane and fin	18·5 kW	6 × 15·8 kW	Cycled with propellers.
Propellers ..		4 × 12·0 kW	Cycled with tail unit.
Four engine intakes ..	28·0 kW		Dry anti-icing.
Four spinners	12·0 kW		Wet anti-icing.
	85·0 kW	48·6 kW max.	Totals
	Maximum total input—133·6 kW		

In the case of a single engined turbo-jet fighter, anti-icing protection will normally be required on the engine intakes only, with probably a small protected panel on the windscreen. The reason for this protection is that, for an aircraft of this type, continuous flight in icing conditions is not required, and thus if icing conditions are encountered, action can be taken to fly clear of them quickly enough for the less critical flying surfaces not to become dangerously affected. However, thrust must be maintained at all costs to enable this evasive action to be carried out, and hence protection for the more critical power unit must be provided, particularly in view of the relatively high rates of accretion associated with high aircraft speed.

Fig. 4.—(Left) The Bristol 'Britannia' fin leading edge fairing with protection on the cyclic de-icing principle, and (right) the Fairey 'Gannet' engine air intake with protection on the continuous dry anti-icing system

Inspection, Maintenance and Repair

The Napier sprayed heater mat has shown itself to be extremely reliable in operation by virtue of the methods used in its construction and application to the aircraft component.

A visual examination is all that is normally required at minor inspection periods. At a major inspection, however, a more extensive visual check is made, and any region showing evidence of mechanical damage is investigated further by insulation and resistance checks employing normal servicing equipment. Where a full ground functioning test is deemed desirable, such a test is carried out at reduced voltage using a special power supply pack.

An important advantage of the heater mat construction is the ease with which electrical repairs may be effected should they ultimately become necessary. Schemes have been evolved whereby such local repairs may be carried out in the workshops without any difficulty, using the hand spraying methods. The damaged section is removed, if necessary, and a new section of base insulation and heater element is then grafted in by spraying, checked for uniformity, and the top insulation applied.

Heater Mat Production

Facilities have been established at the Napier works for the manufacture of the surface heater mat, and its application to complex sections of aircraft and engine components.

Workshop equipment includes recently developed spray guns used in spraying the insulation coats and elements to a wide range of heating intensities. Both manual and mechanical spraying apparatus are included, as these have special applications to both component contour and repair schemes. A number of specially developed multi-purpose spraying machines are installed to facilitate economic production.

To ensure that the sprayed heater mat is of high quality and conforms rigidly to the design requirements, specially developed equipment is used for checking the heater during manufacture, and a series of exhaustive tests are carried out by the production and inspection staffs in the production stages and on the completed product.

The Napier surface heater mat is installed on components of some of Britain's leading aircraft (Fig. 4) including the Bristol 'Britannia' and the De Havilland 'Comet'; also the engine intakes of both the Fairey 'Gannet' and the Short 'Seamew' are protected by this system.

The Hydro-Electric Industry

By CHARLES JAEGER, Dr. es Sc. tech., Consulting Engineer to the Hydraulic Department of The English Electric Company, and Special Lecturer at the Imperial College of Science and Technology, London.

This article is based on the lecture given by the author at the Centre de Recherches Industrielles in Geneva on 22nd March, 1954.

ELECTRIC POWER CAN be obtained by using the energy of steam or by transforming the energy of flowing water. In this article the hydro-power side of the electrical industry will mainly be considered. The aim of the Hydro-power Industry is to produce, transmit and distribute electric energy, generated from hydraulic energy, in order to meet growing demand from the consumers.

The exploitation of hydro-power will depend largely on natural conditions of climate, rainfall, availability of water all the year round and on the existence of a head or difference between the water levels upstream and downstream of a practicable site for the turbines. The cost of hydro-power will depend also on the geological conditions, which the civil engineers have to consider carefully when designing and planning the construction.

Hydro-power depends to a large extent on existing demographic conditions. It may happen that the water-power potential occurs in densely populated areas, as in the Rhine Valley between Basle and Strassbourg or on the Rhône between Geneva and Donzère. More recently, hydro-power generated in poorly populated areas like the Italian, French or Swiss Alps, in the east and north-eastern parts of Canada, or in the far north of Sweden, has to be transmitted hundreds of miles to the more densely populated and industrialised areas. Another unexpected aspect of the later development of hydro-power resources is its force of appeal for new immigration. At a very early stage, the first hydro-power developments in the French Dauphiné contributed to the demographical and industrial development of this French province. More recently, the development of hydro-power in the Tennessee Valley and in the north-west of the United States has caused migration towards these provinces. One of the main objects of the North of Scotland Hydro-Electric Board is to make electrical energy available to the northern parts of Scotland, to encourage new local industries and to counteract the trend towards depopulation which has menaced the northern counties for more than a century. All these aspects affecting the production, the transmission and the distribution of hydro-electric energy will have to be considered.

I. EARLY DEVELOPMENT OF HYDRO-POWER

In France, Aristide Bergès (1833-1904) is accepted as one of the pioneers of hydro-power. In 1869 he equipped the first hydro-power station with a head of 200 metres, an extremely high head for that time. The power produced, about 3.4 h.p., was not electrical but was used directly as mechanical power in a paper mill. A few years before (1867), the first dynamo-machines had been built by Siemens and by Wheatstone. By 1889, that is over a period of twenty years, Aristide Bergès had equipped 17,000 h.p. of plant in the Dauphiné around Grenoble. Meanwhile, Hippolyte Fontaine had improved the Gramme machine, and Marcel Deprez realised the first long distance transmission of energy in 1881 at the Paris International Exhibition and again in 1882 at the Exhibition at München, Germany. In 1883, 7 h.p. was transmitted over a distance of 14 km at Grenoble with an efficiency of 62%. In September 1882, Thomas Edison put into service the first electric generating plant in New York, the generator being driven by steam. A few days later, a generator driven by water power began operation at Appleton, U.S.A. In the year 1885, C. E. L. Brown, founder of the firm Brown Boveri, then at Oerlikon, transmitted 50 h.p. at 1,250 V over a length of 8 km with an efficiency factor of 75%. The first American example of long distance transmission

was in 1889 between Oregon and Portland (450 kW over 13 miles at 400 V). This was the signal for the first considerable hydro-power developments in Europe, and also very soon in the United States.

In the years 1882 to 1885, the first manufacturers of electrical machinery started their creative work in both continents. The year 1884 was of great importance for the introduction of alternating-current machinery which provided the foundation for the electrical industry.

The first hydro-electric power stations were designed and built about 1890, that at Chèvres, near Geneva, being dated 1896.

II. BASIC TECHNICAL ASPECTS OF HYDRO-POWER PRODUCTION

The object of the hydro-power industry is to produce electricity by utilising the energy of falling water. A hydraulic machine or turbine will produce a power N by using a discharge Q over a head H, or better, a net head H_1. The general formula which applies is

$$N = w\,\eta Q H_1$$

where 'w' is the specific weight of the water, and 'η' is an efficiency factor indicating how much electrical energy can be recovered from the hydraulic energy; the quantity $(1-\eta)QH_1$ represents the losses in the turbine and generator.

In this formula all the quantities are in metres and kilograms per second, or in feet and pounds per second. The usual unit of power is the horse-power (h.p.), the normal electrical unit is the kilo-watt (kW) and a power of 1 kW used during one hour represents a unit quantity of energy or 1 kWh. There are conversion factors to transform kilogram-metres or foot-pounds into h.p. or into kW.

The usual method of developing hydro-power is to dam a river and divert the water over a system of open canals or pressure conduits and pipe lines to a power house, where turbines absorb the discharge Q under a pressure head H, thus producing a power N. The water returns to the river through a tail-race canal.

According to the head available, engineers distinguish:—

(*a*) High head power stations equipped with impulse or Pelton wheels;

(*b*) Medium head power stations equipped with reaction or Francis turbines;

(*c*) Low head power stations equipped with feathering propeller or Kaplan turbines.

The basic principles are always the same. If the topographical conditions are favourable, the length of the head-race canals or pressure conduits may be reduced. There are power stations without head-race canal or conduit, and without any tail-race. A typical case is that of a large river being crossed by a dam or weir into which a power station is built. The water is held back by the weir or dam, thus creating the necessary head for the turbines.

The discharge of a river depends on the rainfall, or sometimes on the melting of snow or ice. The discharge varies widely with the season, and the ratio of flood water to low water can be greater than a hundred to one. On the other hand, the demand for hydro-power is fairly constant all the year round, with possibly a slightly greater demand during the winter than during the summer. One of the main problems the hydro-power industry has to solve is the adjustment of the production of the energy to the actual demand. Plants where no storage of water is possible and producing seasonal energy only are called 'run of the river' stations. The aim is to produce firm power, and different means have been worked out for spreading the production of electricity more evenly over the year, by:—

(*a*) creating large reservoirs where water is stored for the dry season;

(*b*) diverting water from one catchment area to another;

(*c*) combining 'run of river' plants with large storage plants (the Tignes reservoir will work in series with Isère Arc, Mauvoisin and Grande Dixence storage in parallel with seasonal power plants on the Rhine—see also the Tennessee Valley Authority's system);

(*d*) using excess energy produced during the wet period, or at night, to pump water into storage reservoirs (so-called pumped storage plants);

(*e*) combining 'run of the river' plants with pumped storage schemes (Kembs on the Rhine, and the Lac Blanc-Lac Noir pumped storage scheme);

(*f*) combining two different 'grids' (the French Alpine hydro-power system with the group of power stations in the Massif Central);

(g) combining hydro power with steam power (in France, half of the production is hydro power and half steam power);

Finally, we must not forget the possibility of developing tidal power, where this power which varies over the day with the tide could be absorbed by a very large 'grid'.

III. ADMINISTRATIVE ORGANISATION OF THE HYDRO-ELECTRIC INDUSTRY

Private enterprise started the whole electrical industry, and private initiative also founded the hydro-power industry. In a paper on the early development of the hydro-power industry in Switzerland, the author[1] underlined the fact that in Switzerland private initiative has always done the pioneering work. The same economical process has also obtained in other countries.

Very soon, cities and counties began developing their own hydro-power stations to meet the increasing demand for this latest form of energy. From the earliest stages, state-owned enterprises became competitors with private concerns.

If to-day we consider more generally the hydro-power industry from the angle of production and distribution of electricity, we find a wide variety of administrative arrangements. In some countries, the industry for the most part is privately owned. This is the case in Italy where private companies (like the Edison Group, the Società Adriatica di Elettricità, and the Montecatini Group) are responsible for the bulk of the national energy production, and are far more powerful than public corporations like the Electricity Board of Milan.

The opposite course was taken in France where, since 1946, the entire electric power industry has been nationalised. Two powerful state-owned corporations govern the whole business of electricity production and distribution. The Compagnie Nationale du Rhône (Authorised by the law of 1921, but established only in 1933) is in charge of power development in the Rhône valley, including irrigation and navigation on the Rhône river. The rest of the country is provided with energy by Electricité de France who took over from private industry after the end of the last war. The arguments in favour of nationalisation of the French power industry, as a counterpart of the nationalisation of the French railways and coalmines, were mainly technical and financial. It is alleged that French private industry would not have been able to raise the money and provide the technical and specialised staff which was required to face the heavy burden of post-war development, as outlined by the well-known Plan Monnet.

Problems of finance are outside the scope of this article. Concerning the technical staff of Electricité de France, this corporation inherited all the staff from the concerns which were nationalised and it is likely that, at the beginning of its activity, Electricité de France was rather overstaffed.

The practical achievements of Electricité de France in the last seven years are considerable. At the peak of the development programme in the year 1950, Electricité de France had accomplished in 10 months the movement of 840,000 m^3 of earth and the placing of 1,260,000 m^3 of concrete. In 1949, this authority excavated 84 km of tunnel. It is possible that many internal changes in French private industry would have been necessary to enable it to face such a construction programme.

The Swiss hydro-power industry is characterised by a high degree of collaboration between private and state-owned enterprises, thus demonstrating what can be achieved when nationalisation is rejected by public opinion but a high degree of control by the state is nevertheless considered desirable.

There are four types of state-owned electricity corporations in Switzerland:—

(1) Electricity boards entirely in the hands of one township, like Zurich or Basle.

(2) Electricity boards directed by a cantonal authority such as the Bernische Kraftwerke (a canton being the equivalent of an English county). Examples are the Electricity Boards of the Canton of Berne and of the Canton of Fribourg.

(3) The powerful Nord-Ost Schweizerische Kraftwerke (N.O.K.) is a corporation grouping several cantons in the north-east part of Switzerland. This N.O.K. group, in spite of being entirely state-owned, has the legal

[1] *Ch. Jaeger. 'Die Entwicklung der Hydraulik und der hydraulischen Wissenschaft in der Schweiz in den letzten 100 Jahren'. Wasser u. Energiewirtschaft, Vol. 40, Nos. 9, 10, 11, Sept.-Nov. 1940.*

status of a private company, all the shares being in the hands of the cantons. A large amount of private money is invested in the N.O.K. corporation as debenture loans.

(4) Finally, the Swiss Federal Railways are the owners of a series of powerful stations in different parts of the country.

Associated with these public corporations, there are different types of private electricity companies. The powerful Aare-Tessin which transmits power in the north-south direction over the Swiss Alps is one of them. The aluminium industry and some chemical companies are also owners of large hydro-power stations. Several companies with the legal status of a private company are really state owned, all the shares being in the hands of cantons or townships (*e.g.* Oberhasliwerke).

To understand how the whole industry works and expands, it is essential to mention the activities of the technical banks, by far the more important of these being Electro Watt (former Banque pour Entreprises Electriques, Zurich) and Motor Columbus.

The Limited Company Motor was founded in 1895 by Walter Boveri, the founder of Brown Boveri and Co. (1891). This firm was responsible for the design and construction of many of the earliest hydro-power stations in Switzerland. Most of these stations were transferred to private or publicly owned power companies, including the Kraftwerke Beznau-Löntsch (1907) and the Officine Elettriche Ticinesi (1917). In 1911 it helped to create the Compania Italo-Argentina de Electricidad in Buenos Aires. After World War I, 'Motor' suffered considerably owing to the post-war currency trouble, which struck its foreign assets, and in 1923 it merged with a subsidiary, the financial trust called Columbus (created in 1913). They had a joint capital of 60,000,000 Swiss francs which was later increased to 93,000,000 frs but went down to 55,250,000 frs in 1938 as a consequence of the economic crisis. To-day, the capital is back to 60,000,000 frs. To alleviate the balance sheet, the debentures were reduced successively from 82,000,000 to 37,000,000 francs during the years 1938 to 1945. Since this re-organisation, Motor Columbus has been engaged successfully on consulting engineering work and in studying, organising, partly financing and launching a series of large hydro-power schemes. It was responsible for:—

Ryburg Schwörstadt on the Rhine (1929-31), 108,000 kW;

Piottino on the Ticino (1931);

the first power transmission line in the south-north direction over the Gotthard Pass (1933);

Lucendro near the Gotthard Pass (1945-47), 48,000 kW.

To-day Motor Columbus is participating actively in the development of the enormous Grande Dixence Scheme.

The story of Electro Watt, former Banque pour Entreprises Electriques, is very similar. This group was founded as a technical bank with the purpose of financing hydro-power projects directly, or of helping the promotors in raising the necessary capital.

The two World Wars and the economic crisis between the wars were highly detrimental to Electro Watt, owing partly to its extended assets in foreign countries. Since the end of World War II, the firm has been active again and is now concerned with the recent developments of the powerful Mauvoisin scheme.

In the period following the end of the war in 1945, both groups have fully recovered financially. They are now concentrating on acting as technical and financial advisers to large power schemes inside Switzerland. Recently, both groups resumed a limited activity on the international market also.

The picture of the Swiss hydro-power industry past and present would not be understandable without a clear description of how these two groups work, being strong competitors one to the other, but collaborating intimately with private and with public producers of hydro-electricity. From the financial side their activity is of great importance, as it backs and maintains the independence of private industry and gives powerful backing to the Swiss Stock Exchange and to private capital which finds in the hydro-power industry a desirable opening for large investments.

This section on the administrative aspects of the

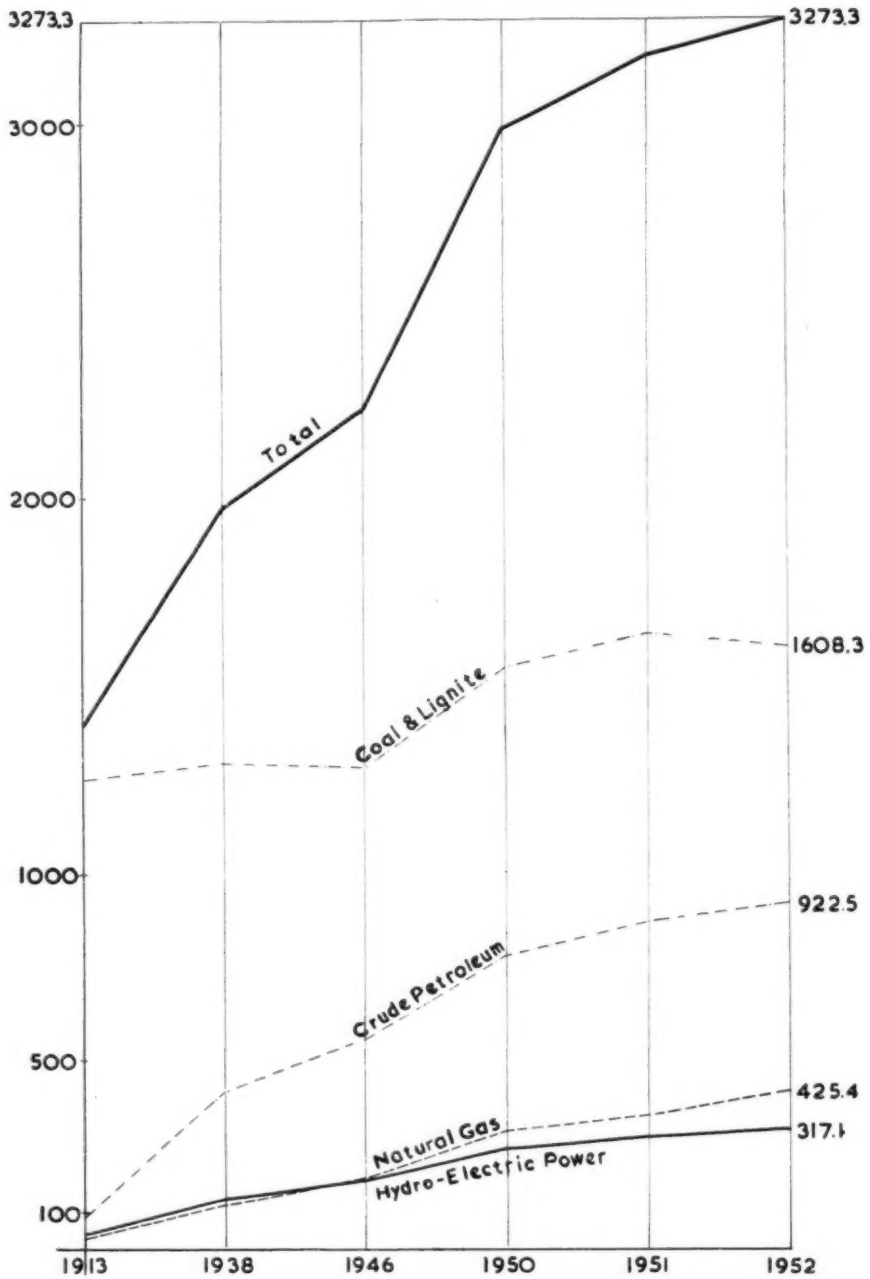


Fig. 1. World production from the main power sources in coal equivalent in millions of metric tons (1913-1952)

hydro-power industry may be summarised as follows:—

Private initiative started the whole hydro-power industry and in several countries still holds the whole industry firmly in its hands, or shares its control with state-owned and state-controlled bodies.

The nationalisation of the hydro-power industry, or in more general terms the electric power industry, occurred very largely

(*a*) when a nation had to develop its hydro power rapidly with little previous experience (*e.g.* India), or

(*b*) when difficult economic conditions urgently required standardisation of existing industries and their rapid extension. These were the conditions which forced nationalisation upon the British and the French electrical industries immediately after World War II.

It is increasingly important to entrust to trained engineers the administrative and management responsibilities at a high level, if the hydro-power industry is to be prosperous. The financial success of the industry depends to a large extent on the technical ability of the staff, and this aspect is worthy of further discussion concerning the technical staff, their training and technical ability, and the relations of the hydro-power industry with its technical consultants.

IV. THE PRESENT PRODUCTION OF HYDRO-POWER IN RELATION TO MAIN SOURCES OF ENERGY

The production of hydro-power in the world compared with the world production from other main power sources is best shown in a series of tables. The first (Table I) refers to the world power production between 1913-1952.

For comparison, these figures have been translated into coal equivalent measured in millions of metric tons in Table II, and are shown graphically in Fig. 1.

TABLE I

World Production of Power from the Main Sources of Supply

Year	Coal and Lignite, millions of metric tons	Crude Petroleum, millions of metric tons	Natural Gas, 1,000 millions of cubic metres	Electric Power, 1,000 million kWh	
				Total	Hydro-electric
1913	1,256·1	53·7	18·1	—	40
1938	1,300·5	280·7	78·5	460·6	172·9
1946	1,290·9	376·8	132·4	644·0	232·0
1950	1,551·3	524	212·2	951·4	332·7
1951	1,640·3	592	239·7	1,060·5	368
1952	1,608·3	618·1	284·2	1,134·0	397·0

TABLE II

World Production of Power from the Main Sources, given in Coal Equivalent in millions of metric tons

Year	Coal and Lignite	Crude Petroleum	Natural Gas	Hydro-electric	Total
1913	1,256·1	80·1	27·0	32·0	1,395·2
1938	1,300·5	418·9	117·2	138·3	1,974·9
1946	1,290·9	562·4	197·6	185·6	2,236·5
1950	1,551·3	782·1	316·7	266·2	2,916·3
1951	1,640·3	877·1	357·9	294·5	3,169·8
1952	1,608·3	922·5	425·4	317·1	3,273·3
			in percentages		
1938	65·9	21·2	5·9	7	100
1952	49·1	28·2	13·0	9·7	100

TABLE III

Annual Rate of Increase in the World Production of Power

Period	Coal and Lignite	Crude Petroleum	Natural Gas	Hydro-electric	Total
	%	%	%	%	%
1929-1938	−0·75	3·50	3·50	5·00	0·50
1938-1946	−0·10	3·38	6·75	3·75	1·60
1946-1952	3·75	8·60	13·60	9·30	6·56
1920-1952	0·82	5·88	8·20	6·35	2·55

Based on these figures, it is possible to calculate the annual rate of increase in the world production of power, as given in Table III and Fig. 2.

For the period 1938-1952, the average annual rate of increase in the production of electricity has been as follows (Table IV).

TABLE IV

Countries	%
Austria	7·3
France	5·7
West Germany ..	17·1 (1947-52)
East Germany ..	9·6 (1947-52)
United Kingdom ..	7·1
Italy	5·1
Norway	4·7
Portugal	8·6
Spain	8·4
Sweden	6·9
Switzerland ..	4·5
Europe	4·8 (without U.S.S.R.)
U.S.S.R.	8·0 (1921-38)
U.S.A.	8·6
Canada	6·3
Mexico	5·7
Argentina	5·2
Brazil	7·5
Chile	9·9
Columbia	10·3
Venezuela	14·5
India	8·7
Australia	7·3
New Zealand ..	6·8

The present trend of power increase is in most countries even more energetic.

In the year 1952, the production of hydro electricity on the different continents was as follows (Table V).

TABLE V

Production of Hydro-electric Power in 1,000 millions of kWh for the Year 1952

Europe	136
U.S.S.R.	13
Asia	52·5
Africa	1·8
North America ..	175
South America ..	13
Australia	5·7
Total	397·0

This is out of a total of 1,137 $\times$ 10^9 kWh, including thermal energy, for the same year 1952.

The production of hydro-electric power for the more important countries has been as in Table VI.

TABLE VI

Total Production and Hydro-electric Production of Power for the Year 1952 in 1,000 millions of kWh

Countries	Hydro	Total
Austria	6·36	8·02
France	22·45	40·8
West Germany ..	9·946	56·2
East Germany	—	21·7
United Kingdom ..	1·27	61·99
Italy	27·1	30·84
Norway	18·67	18·77
Portugal	1·2	1·33
Spain	6·95	8·45
Sweden	19·5	20·7
Switzerland	13·0	13·1
U.S.A.	109·6	462·6

Canada	59·5	61·7
Mexico	—	5·3
Argentina	—	4·7
Brazil	—	3·08
South Africa	—	12·5
Japan	40·7	50·88
Turkey	—	1·07
Australia	5·7	11·7
New Zealand ..	3·4	3·5
U.S.S.R.	—	117·0

For the foregoing figures the sources of information are the publications of the United Nations Organisation and the U.N. Economic Committee for Europe. The Italian paper Quaderni, Volume 9, No. 147, May 1953, was also consulted.[2]

V. ASSESSMENT OF THE WATER POWER AVAILABLE

Having described the present utilisation of hydro power as compared with other sources of energy all over the world, and having ascertained the general trend of this utilisation, the interest may be shifted to the future of the hydro-power industry. There must be somewhere a limit to the possible development of water power, depending on natural conditions of climate, rainfall, the gradient of the rivers, etc. The next step in this enquiry will be to assess the hydro-power available and to compare the present output with the potential output.

Estimation of the available water power is now-a-days considered a matter of national importance and in most countries it is done on a national basis, even in those countries where the production of hydro electricity is not nationalised. Very recently, the Economic Commission for Europe Committee on Electric Power of the United Nations published an extensive report on 'Hydro-electric Potential in Europe and its Gross Technical and Economic Limits' (Geneva 1953). This report will prove most valuable when estimating possible transfers of energy from one country to another on the European Continent. The enquiry revealed that there are some economic or physical constant values which can be transferred from one area to another, from one country to another. The report claims that the knowledge of these constants, as obtained by a detailed analysis of the conditions in countries where the hydro-power industry has reached a high level of development, may be of great assistance to

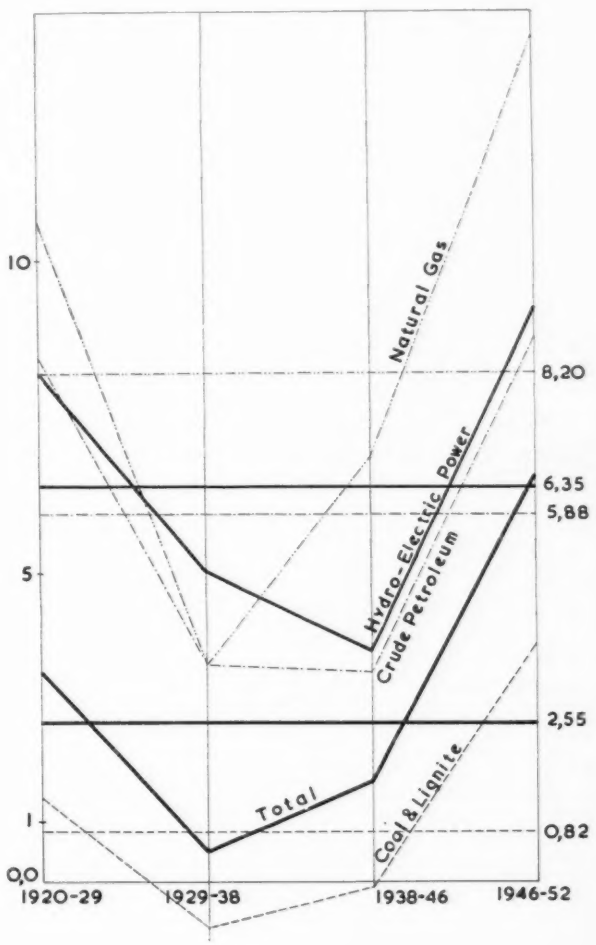


Fig. 2. Percentage annual rate of increase in the world's power production (1920-1952)

[2] *See also F. Bolens, Bulletin Technique Suisse Romande, Vol. 79, No. 11-12, June 13, 1953.*

authorities charged with assessing the water power available in under-developed or newly prospected countries.

In the introduction to the Report we are told that "It is possible to assess the water-power resources of different regions on a consistent basis if three alternative levels of definition can be used. These levels may be stated as follows:—

(*a*) The gross potential power of the head and flow of water physically available in an average year.

(*b*) The proportion of this latent power which it is technically possible to harness at a given time.

(*c*) The proportion which can be exploited economically at a given time."

The importance of introducing these clear definitions is best shown by Table VII, an official estimate of hydro resources in Switzerland, which shows how the estimates may vary when knowledge of the physical and climatological data involved becomes greater.

practice, largely sufficed to cancel out the opposing tendency for real costs to rise parallel to rises in wages and salaries.

Wide variation in the estimates of available hydro-power made it necessary to give clear definitions of the gross, the technical and the economic possibilities.

The yearly gross hydro-electric potential is given by the expressions

$$\frac{1{,}000}{75} \cdot Qh \cdot 0.736 = 9.81Qh \text{ kW, or}$$

$$8{,}584Qh \text{ kWh}$$

assuming that the generating stations run during the 8,760 hours of the year,

where Q = arithmetic mean over a period of years of the yearly average natural flows (m^3/sec) at the mid-point of the section of the water course considered, and h = gross head in metres, *i.e.* difference in level between the ends of the section considered.

TABLE VII

Official Estimates of Hydro Resources in Switzerland (1898-1946)

Year	Q95* MW	Q50* MW	Arithmetic mean flow MW	Mean annual production possibilities 10^6 kWh	Actual annual production 10^6 kWh	Col. 6 as percentage of col. 5 %
1	2	3	4	5	6	7
1898		470		4,500		
1914	660	1,880		15,000	1,300	9
1934	2,400	3,800	5,100	20,000	5,312	27
1946	3,300	5,200	7,000	27,000	10,060	37

* *The flow available for 95% and 50% of the time.*

Trends in assessment and exploitation such as demonstrated in this table are common to practically all countries possessing hydro resources in a relatively advanced stage of development. They have arisen, first, from improvements in the assessment of physical features of sites, from accumulation of fuller and more reliable records of precipitation and run-off, and from more complete topographic surveys. Next, the conception of what is technically possible within acceptable limits has been widened enormously in recent years, thus causing ever greater demand for accurate surveys. During the last 30 years, various technical advances have, in

The basic formula then gives the maximum yearly potential for a given flow through a given difference of level: it assumes:—

Complete equalisation of seasonal flow;
complete utilisation of the head;
100 per cent plant efficiency;
100 per cent utilisation of the energy.

Methods have been developed in France, Italy and Switzerland for assessing correctly the mean flow Q and for integrating correctly the values Qh over large areas. The Electric Power Division of the Economic Commission for Europe has developed

its own method, which has been compared with others. The estimation of the gross power is a difficult problem in countries where accurate data on the hydrology of the main rivers have not been accumulated over long periods of years. It is a far more formidable task in under-developed countries or newly prospected countries where such data do not exist at all.

The determination of the technical potential is a completely different task which is based on detailed planning of future power stations in order to assess the output potential. The technical skill of the planner may influence the final estimate. In addition to this personal factor, the final result is influenced very greatly by evolution of design technique. The new techniques developed for the design of very high dams, the new ideas for extensive diversion of water from one catchment area to another and, finally, the new technique for underground stations allow a far better utilisation of the water power over large areas and therefore provide a higher estimate.

(*d*) Changes in interest rates.

(*e*) Prospects for the large-scale utilisation of new sources of power (tidal projects, nuclear energy, etc.).

In addition, in many regions the possibility should be taken into account of more and more accurate surveys and of surveys in areas formerly disregarded. The possibility of transmitting electric energy at ever higher voltages over ever increasing distances may influence profoundly the future economics of water power. Mention should be made here of the existing long-distance transmissions in Sweden and the U.S.S.R., and of the future development of the Canadian hydro-power system which will require similar long-distance transmission.

In all European countries the estimates for gross, technical and economic potentials show a steady increase, which testifies not only to the advance of technique but also to the ever increasing pressure of demand from the consumers for more power.

TABLE VIII

Estimated National Gross, Technical and Economic Hydro Potential in 1950

Country	Gross yearly potential at full utilisation		Maximum production technically possible		Output from projects considered economically possible		
	10^6kWh	kWh/ha	10^6kWh	Col. 4 as percentage of col. 2	10^6kWh	kWh/ha	Col. 6 as percentage of col. 2
1	2	3	4	5	6	7	8
Switzerland ..	144,000	36,000	70,000	48·6	27,000	6,750	18·7
Austria	152,500	18,200			30,000	3,580	19·7
France	314,000	5,710	138,000	44·0	60,000	1,087	19·2
Germany (E. and W.)	111,000	3,250	26,000	23·4	18,000	510	16·5
Czechoslovakia ..	39,300	3,080			7,000	548	17·8

The estimate of economic potential depends on further factors:—

(*a*) Technical progress as it affects both specific production costs of conventional hydro-electric and thermal operating plants, and the range of hydro-electric civil construction that is practicable.

(*b*) The rate of increase in demand for electricity.

(*c*) The costs of the various fuels for thermal plants.

The main results from the careful investigation by the Power Division Economic Commission for Europe are summarised in the above table (VIII).

The figures in column 5 are of general importance, as they vary inside narrow limits and permit a rough estimate of the power technically possible in an under-developed country where only rough figures of the gross yearly potential are available. At the present stage of hydro-power technique it is not possible to utilise more than half the gross yearly potential, and the economically possible

TABLE IX

Economic Hydro Potential in Europe in 1950

Country	Area	Estimated yearly hydro production economically exploitable		Already developed, as percentage of total resources	Still to be developed	
	$10^3 km^2$	10^6kWh per annum	kWh/ha		10^6kWh per annum	kWh/ha
1	2	3	4	5	6	7
Austria	83·8	30,000	3,580	17	25,025	2,990
Belgium	30·5	545	170	12	480	150
Bulgaria	110·8	6,000	540	6	5,630	510
Czechoslovakia	127·8	7,000	550	16	5,900	460
Finland	303·9	10,000	330	36	6,360	210
France	550·3	59,800	1,090	27	43,610	790
Germany (E. and W.)	353·2	18,000	570	50	9,000	250
Greece	113·6	4,260	370	—	4,250	370
Hungary	93·3	1,500	160	4	1,440	160
Ireland	68·9	1,050	150	44	600	90
Italy	301·0	50,000	1,660	43	28,400	940
Norway	323·8	104,500	3,230	17	86,845	2,680
Poland	310·1	5,500	180	9	5,000	160
Portugal	88·8	7,970	900	5	7,530	850
Roumania	237·5	21,600	910	1	21,400	901
Spain	492·4	31,500	640	16	26,500	540
Sweden	408·5	50,000	1,220	35	32,520	800
Switzerland	40·0	27,000	6,750	38	16,680	4,160
United Kingdom	241·7	9,400	390	20	7,525	310
Yugoslavia	256·6	50,000	1,950	2	48,900	1,910
Turkey	767·1	15,000	190	—	14,970	190

output is usually less than 20 per cent of the estimates for gross power.[3]

On the basis of this survey the Electric-power Section[4] has published a formula for Europe, relating the density of gross potential 'd' in kW/km^2 to the percentage 'p' of that gross power which is at present technically installable. The formula is

$$p = 0{\cdot}103d + 9$$

The value 'd' depends on the altitude and geographical position of the area. It is equal to 443 for the upper Rhône in Switzerland at 1,340 m average level and is equal to 179 for the Durance Verdon area in France at the level 1,040 m.

It is of great interest to compare the present developed hydro power with the estimated yearly hydro-power production economically exploitable. Table IX gives the figures for the year 1950 for European countries.

The year 1950 was the occasion of the Fourth World Power Conference in London, when many data from countries all over the world were made available. The figures published by the Economic Commission can now be supplemented from other countries. In the United States in 1949, 87,000 million kWh were produced annually by water power generation. In 1952, 109,600 million kWh were produced by water power out of a total of about 462,000 million kWh (steam and hydro) and an estimated hydro-power potential of $500{,}000 \times 10^6$ kWh, but these figures are not arrived at on the same basis as those published for Europe. According to reports submitted to the World Power Conference (1950), and others made available more recently, the figures in Table IXa are valid for Canada.

The estimates in this table are based on the actual head or drop at existing falls or rapids or known power sites. No consideration is given to possible economic concentration of head. The increase in the figures for available power between the years 1950 and 1952 is due to extended surveys of rivers. Many great rivers in Canada have never

[3] *In a very recent survey by Electricité de France, a figure of 29% is reached for France. The French survey is based on detailed studies of actual projects, the technical potential and the economic potential being assumed identical. See M. Rousselier, Directeur adjoint de l'Equipement à E.d.F. 'L'Inventaire des Ressources Hydroélectriques potentielles françaises', Travaux, Vol. 38, No. 240, October 1954, pp. 737-744.*

[4] *E.C.E. 1953 Report, pp.* 81-83.

been surveyed in detail. For India, the figures published at the World Power Conference are 25×10^6 kW, equivalent to possibly $220{,}000 \times 10^6$ kWh for ultimate water-power potential. A systematic and comprehensive survey of the water-power potential of India has not yet been undertaken, since, says the report, the cost of such a survey would be out of proportion to its practical value. The potential continuous output of sites already investigated in India amounts to $8{\cdot}5 \times 10^6$ kW, while the installed capacity in 1948 was 576,000 kW with an output of $2{,}372 \times 10^6$ kWh.

Figures which have been published giving the overall potential of China, Africa and other countries of the world must be treated with great caution. Let us admit that we are here largely ignorant and can do little more than make an intelligent guess.

In a few cases, tidal power may add substantially to the water-power potential. It has been calculated that the Severn tidal scheme near Bristol would yield $2{,}000 \times 10^6$ kWh a year, and the St. Michel tidal scheme in France up to $20{,}000 \times 10^6$ kWh annually.

Recently, a realistic estimate of the world's water power as at the end of 1952 was prepared tentatively by another author and is reproduced here as Table X. The figures in column 3 compare with those in Table VI, and the figures in columns 2 and 6 with those in columns 3 and 5 of Table IX. Columns 4 and 5 show that the power is assumed to be used over a period of 4,500 to 5,000 hours a year (corresponding to a load factor of about ·55 which is a fair average) instead of the 8,760 hours used in other computations.

TABLE IXa

Hydro Power in Canada, in Kilowatts

Year	Power at 80% efficiency available 24 hours		Developed capacity (turbine rating)
	At minimum flow (Q95)	Six months flow (Q50)	
Estimate for 1950	20,000,000	37,600,000	8,299,800
Estimate for end of 1952	29,200,000	50,705,000	14,305,880

TABLE X

Realistic Estimate of the World's Hydro-electric Power, as at the end of 1952

Countries	Energy in kWh $\times 10^6$ p.a.		Power in MW		% Developed
	Ultimate Developable	Developed to end of 1952	Ultimate Developable	Developed to end of 1952	
North America*	807,500	175,000	161,500	32,500	20
Europe (excluding Russia)*	491,100	133,646	98,050	29,625	30
Russia in Europe (say)	90,000	20,000	20,000	5,000	25
Siberia and other U.S.S.R. (say)	111,000	13,200	25,000	3,000	12
Asia (excluding U.S.S.R. and China)	454,000	44,300	88,000	10,280	11·7
China	250,000	2,500	50,000	500	1
Australia*	24,500	3,000	5,560	605	10·8
Papua and New Guinea	66,000	450	15,000	10	0·1
New Zealand*	40,000	3,400	8,000	800	10
Africa	450,000	1,320	100,000	300	0·3
East Indies, Philippines, Pacific Islands, etc. (say) ..	66,000	4,500	15,000	1,000	6·7
South America (say)	450,000	27,000	100,000	6,000	6
TOTALS	3,300,100	428,316	686,110	89,620	13·06

* *The figures for these countries are very tolerably reliable estimates of the realisable power.*

VI. THE FUTURE OF THE HYDRO-POWER INDUSTRY

The future of the hydro-power industry is well worth considering. The task for the next 20 to 50 years will be to fill the gap between the present capacity and output as set out in Section IV and the ultimate capacity or output as forecast in Section V.

It is possible to collect a certain amount of information bearing on the near future of the hydro-power industry, but far more difficult to obtain precise data for the general planning of the industry over a longer period.

Several countries have published lists of the projects now under construction or of those they will be developing in the next two or three years. It is therefore possible to estimate the hydro-power capacity in these countries for the years 1956 and 1957. Table XI gives such a forecast for the end of the year 1956.

7 to 9 per cent per annum for these developments is a fair estimate. Roughly, in the next 10 to 15 years the world's developed water power must be doubled or trebled at least, possibly far more. The energy output may rise to three or four times the present figures. This is a colossal task, as the present standard of production has taken about thirty-five years to attain.

As an example, the following figures for Canada were submitted by Mr. F. L. Lawton of the Canadian Aluminium Company.

Present installed capacity ..	14,700,000 h.p.
Average yearly increase in power demand	7·5 per cent
Probable requirement by 1975	75,000,000 h.p.

Figures like these require many changes in the present method of approach and in the capacity for production of plant.

TABLE XI

Estimated Installed Capacity and Output at end of 1956

Country	Installed capacity in 1,000 kW		Output in 1,000 millions kWh
	Hydro-electric	Total	
Austria	2,132	2,888	9·9
France	7,200	13,500	53·4
Italy	10,766	13,244	42·5
West Germany	2,700	15,000	64·5
United Kingdom	700	21,700	73·9
Norway	4,320	4,420	23·4
Sweden	5,050	6,200	26·4
Switzerland	3,700	3,900	15·2

Beyond this very near future, prediction is very difficult. No projects, nor figures, have been published by authoritative sources.

We must base our estimates on the assumption that, for the next ten to fifteen years, the power from all sources will have to be increased by about 6 per cent per annum, with no prospect of the production from coal and lignite increasing more than 3 to 3·5 per cent a year. Water power, possibly along with natural gas and atomic energy, will have to face the bulk of the future increases in demand. Tidal power may have to be considered as part of the future hydro-power resources, and a figure of

There is a feeling among people responsible for the development of water power that long term planning is not given the importance and attention it requires.

For the future, we need to consider the following points.

(*a*) The balance between base load and peak load, between firm power and seasonal power, will probably alter considerably.

(*b*) Very long distance transmission at extra high voltage will have to be adopted. Many of the technical problems involved have still to

be solved. Extensive underwater transmission by cable will become normal practice.

(c) Migration of people with marked displacements of the centres of gravity of the population may occur. Recently, the increased development of water power in the northwest of the U.S.A. has stimulated considerable migration toward this area. Migration of this type will have to be expected in the future. Financial and demographic planning must go ahead parallel to technical developments.

The planning, designing, tendering and contracting, the manufacture of the plant, and the construction of a large hydro-electric scheme ordinarily take about six years; in special circumstances it may be slightly less or considerably more. It is not too early to consider now what will have to be done in 1960.

At that date, huge projects now considered somewhat fantastic, like the Yukon River Project or the Hamilton River Project in Canada, the Saint Michel Tidal project—possibly to be shared by France and Great Britain—and some projects in Africa, may then be close to actuality. In the meantime, important projects like the Snowy Mountains scheme in Australia, the Kafue, Kariba and Volta projects in Africa and many others in the Belgian Congo, India and elsewhere will probably be well under way.

It is highly desirable for the planning of the future to go boldly ahead with designs for new hydro installations, and to engender a greater degree of co-ordination between all the sections of the social organism which are involved.

Quick Starting of Steam Turbines

By P. F. CARSON, A.I.I.A., Senior Technical Service Engineer, Steam Turbine Division.

OVER THE PAST five years the time involved in running a steam turbine up to synchronous speed, and the rate at which electrical load is then applied, have become questions of increasing importance, requiring considerable investigation.

Up to 1939, and immediately following the war years, the need for power in Great Britain was such that the great majority of machines operated continuously to meet base load requirements, and at the periods of peak loading, reductions in system frequency and even load shedding were not unusual.

With an increasing capacity of generating plant as new installations were completed, the load factor has become such that while the peak periods call for the maximum output of installed capacity, the demand during off-peak hours is considerably below capacity, and it is now necessary to reduce load or shut down machines during off-peak periods. From a question of economy, calculated on either the fuel costs or the thermal efficiency of

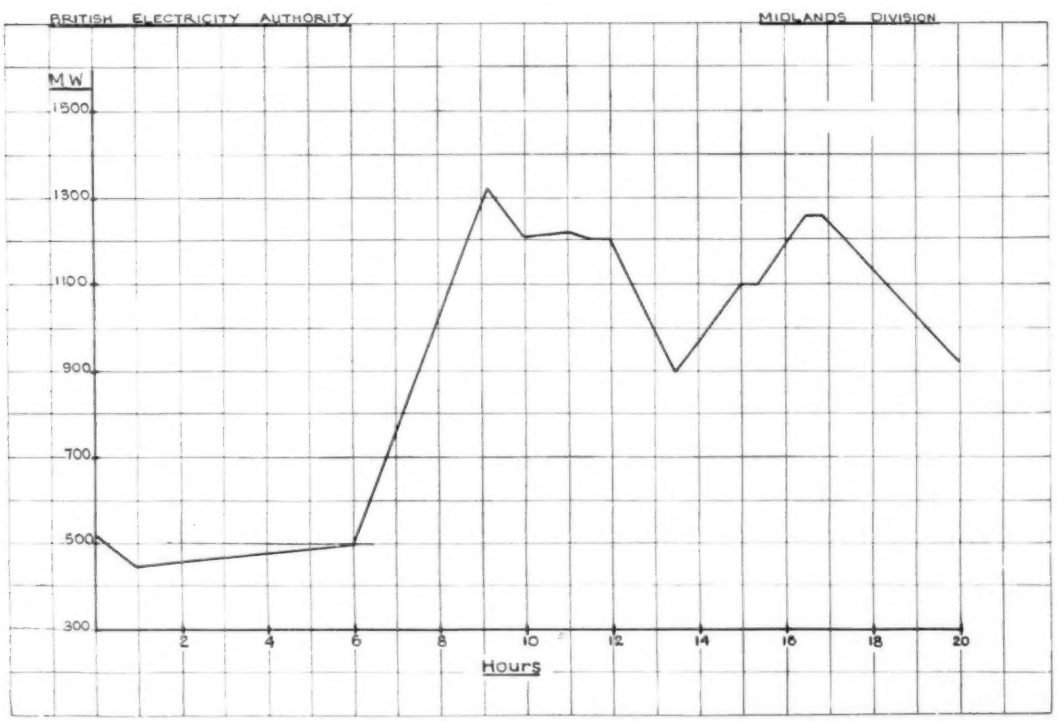


Fig. 1.—A typical winter local daily demand curve for 1951

a particular plant, only the machines of highest efficiency and maximum economy are maintained as base units. Fig. 1 shows a typical winter local demand curve for 1951.

A station operating with high thermal efficiency may, however, suffer from high coal handling costs which reduce seriously the overall economy, and the tendency for the future is to operate units of high efficiency and low fuel cost, i.e. those situated close to the coal producing areas, as base load units, and to shut down at off-peak periods other stations operating at higher costs.

Machines which have been shut down during off-peak hours require to be brought back into service at a speed sufficient to meet the rate of increase of demand.

It is estimated that following completion of the 275 kV grid the peak demands on the London Division will be as high as 5,000 MW between the hours of 6 a.m. and 8 a.m. Figs. 2 and 3 show the load and demand curve for 1953-54 as a comparison with the estimated Load and Generation Programme for 1960-61.

While the rate of individual turbine loading will depend upon the number of units which can be started and loaded simultaneously, it has been considered desirable to conduct tests to prove within the limits of safety and reliability the quickest times of run-up possible, together with the maximum loading rate permissible. These tests have covered both 'unit' and 'on range' installations, with steam conditions ranging from 600 p.s.i.g. 850°F, 900 p.s.i.g. 900°F, to 1,250 p.s.i.g. 975°F.

Factors in Operating Techniques

From the data obtained in all tests certain criteria have been established which govern the speed of run-up and the subsequent loading rate, and consideration of these has established techniques whereby after a five hour shut-down a 50 MW turbine has been supporting full load in as little as 33 minutes after first admitting steam. The factors which determine the operating techniques and overall timing are as follows.

1. *The heat transfer and mass relationship between turbine rotors and casings.*

In a machine with operating conditions of 900 p.s.i.g. 900°F and above, the high pressure casing is of greater mass than the rotor, and the construction of the rotor is such that a greater surface area than that of the casing is steam swept (Fig. 4). It follows that during heating or cooling, the rotor expansion or contraction will be more rapid than that of the casing. In short, there is a time lag before equalising expansion values of the two components are reached. This time lag results in a differential expansion value between them, which increases or decreases the axial running clearances between moving blades and fixed nozzles ; this is illustrated in Fig. 5. Relative expansion is described as positive when the rotor becomes longer with respect to the casing.

2. *The temperature of the casing and the rotor at the moment of start-up.*

The rate of cooling for the casing from the moment of shut-down to the time of start-up is of the order of 25°F per hour (Fig. 6), and although the rotor temperature at the bearings and shaft extremities is relatively low, the temperatures within the gland housings are within 100°F of the metal temperature of the housings. Appreciation of this latter point has led to conditioning the gland steam temperature at starting to a value within 75°F of the casing metal temperatures.

The most favourable starting condition is when the steam temperature is 100°F-150°F above the inlet belt metal temperature, thereby assisting in reducing the effects of differential expansion between rotor and casing.

In some designs of casing construction the question of differential temperature between the casing horizontal joint flange and the joint bolts is of importance. When starting under conditions where the inlet steam temperature is considerably above the metal temperature, the inner surfaces of the flange are heated more rapidly than the bolts, which receive heat by conduction at the nuts and also by radiation across the air gap in the bolt holes. It follows, therefore, that severe temperature differentials between these components could result in greatly increased tensile stresses at the

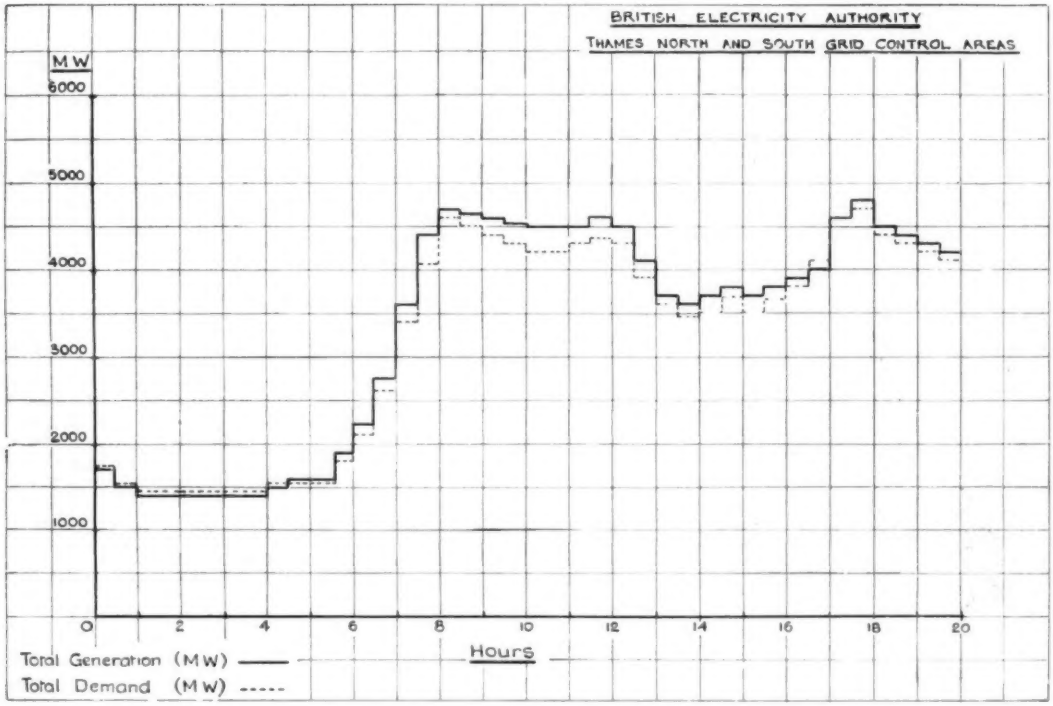


Fig. 2.—Actual winter weekday load and demand for the year 1953-1954

flange. Such conditions could cause crushing of the joint faces and overstressing of bolts. Alternatively, relaxations of stress could result in joint leakage.

This problem has not been encountered in the investigations here concerned, the maximum differential temperature being only 75°F.

The mass of the flange section relative to the casing wall section, and the possibility of introducing high temperature gradients between these parts, is of importance.

Controlled application of flange heating steam reduces the two foregoing effects, and the heating of the flange, and casing, causes a reduction in the temperature time lag between rotor and casing, thereby assisting in reducing the differential expansion values between them (see Fig. 7).

3. The temperature of the steam relative to shaft and casing metal temperature during starting, run-up and subsequent loading.

In circumstances where the turbine has been down for several days, and when the metal is at ambient temperature, the steam at starting may be several hundred degrees above the metal temperature. In a station where the boilers are interconnected and where the turbine is one of several (usually referred to as an ' on range ' system), it follows that at the time of starting any one machine, other machines may be supporting load and therefore the steam temperature available will be at full rating.

When starting a turbine under these circumstances, flange heating steam is applied throughout the whole period between steam admission and the turbine reaching rated speed.

In order to ensure adequate drainage of any

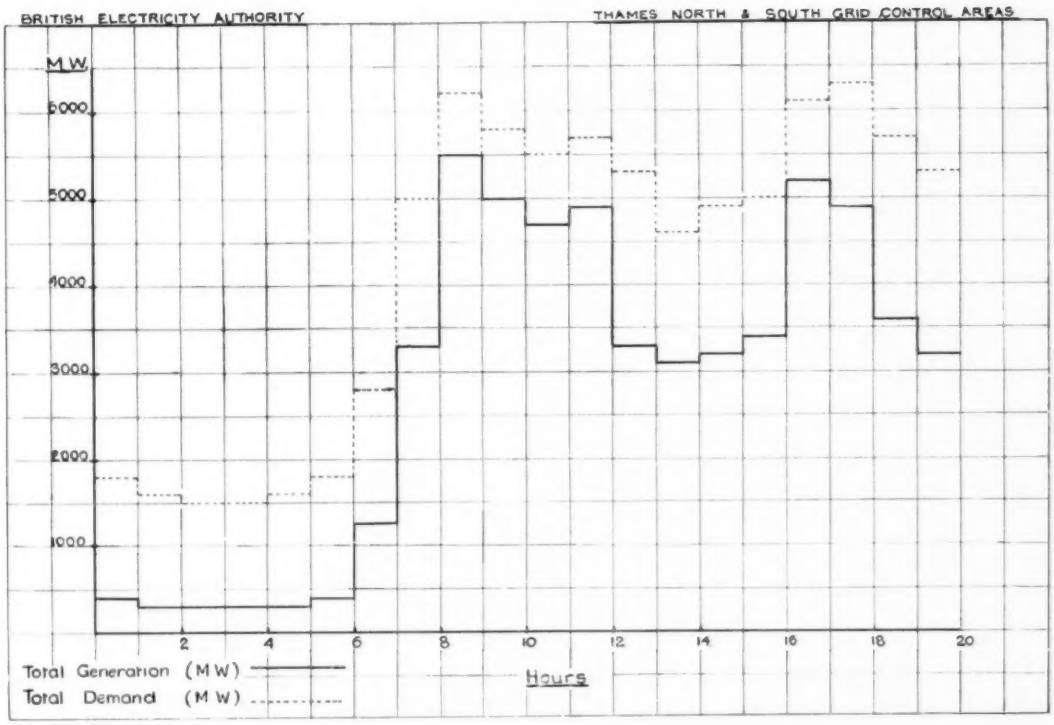


Fig. 3.—Anticipated average load and generation loading programme for the year 1960-1961

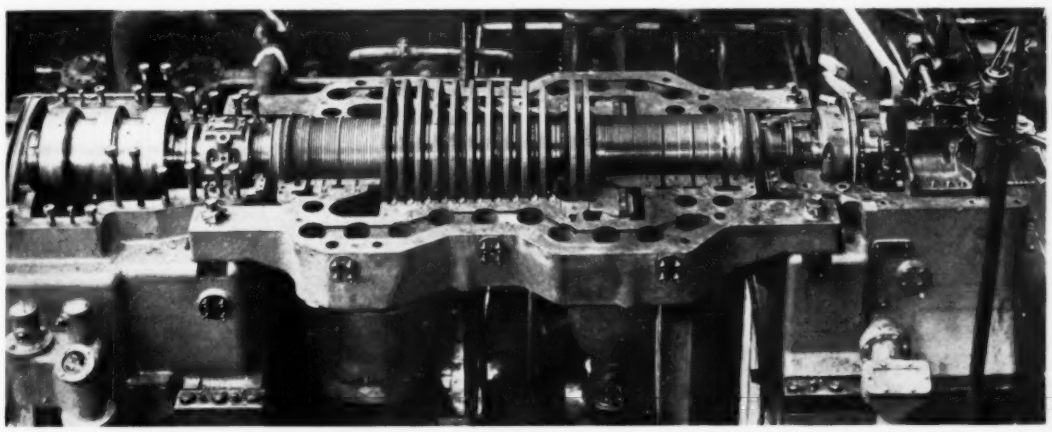


Fig. 4.—Showing the mass relationship between rotor and casing of a high pressure turbine. The flange heating passage is also clearly visible

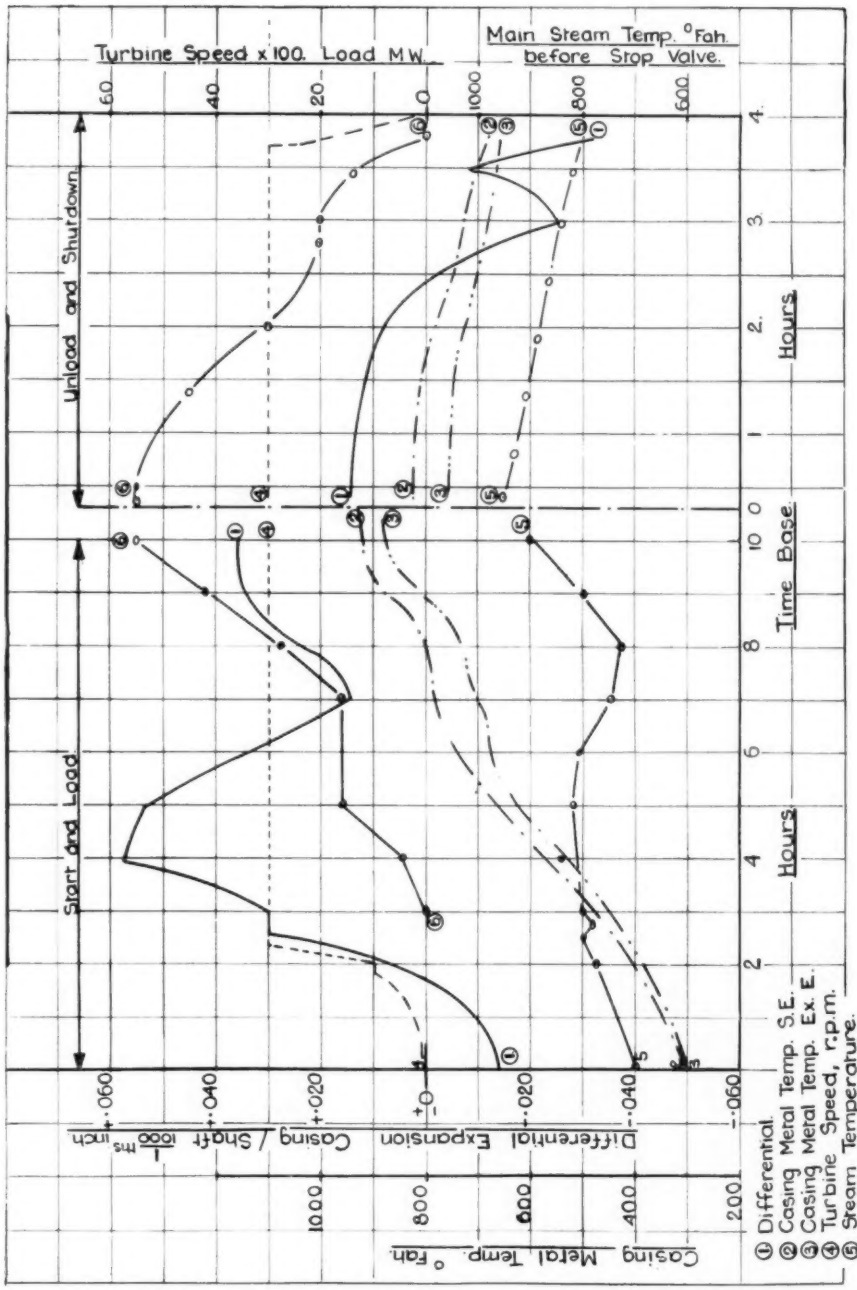


Fig. 5.—Curves of shaft/casing differential expansion

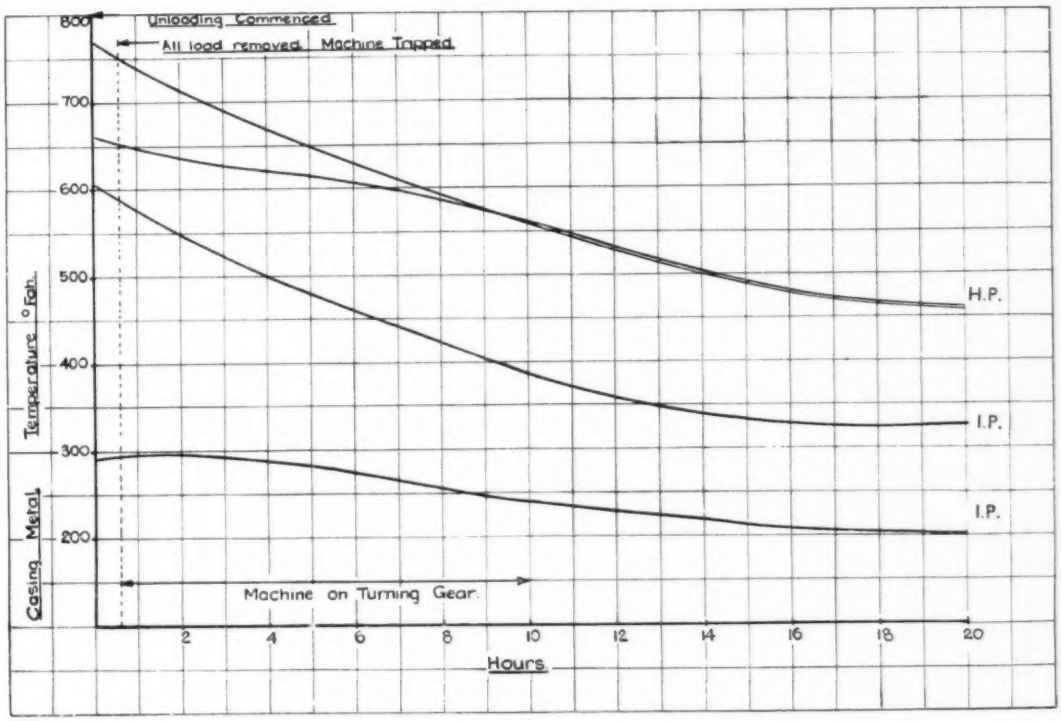


Fig. 6.—Turbine casing cooling curves ; steam condition 600 p.s.i. 800 F

condensation in turbine loop pipes and casings, and also to avoid introducing high thermal gradients within the casings, the rates of application of steam and of speed increases are relatively low when compared with the starting of a hot machine.

Apart from these considerations, the rate of run-up and loading will be controlled by the differential expansion value between rotor and casing.

Where a machine has been shut down for 4 to 8 hours, as under two-shift conditions, the ' on range ' installation has the advantage of having steam available at a temperature above the metal temperatures, and very rapid rates of run-up and loading have been achieved with such conditions. A comparison in initial steam conditions and their effect, observed between ' unit ' and ' on range ' installations, is given in Fig. 8 (a and b).

With *the ' unit ' installation* where one boiler supplies steam directly to the turbine, a start-up from cold presents no serious problem of steam-to-metal temperature differential, the only concern being to ensure that superheated steam is available at the turbine stop valve, and that drainage is adequate. Appreciating that the mass of turbine rotor and casing may comprise only one-third of the total mass which has to be heated—the other components comprising the piping between the boiler outlet and the turbine stop valve, the steam chests, and associated loop piping to the turbine inlet—it follows that drainage is important. Most modern boilers in ' unit ' installations require a steam flow of at least 30% of rated capacity before the steam temperature can be maintained at rated value (see Fig. 9). Elevation of furnace gas temperatures without a steam flow sufficient to control the metal temperatures of superheat tubes can result in failure of this element. Boiler drum tangential and radial stresses must also be kept under observation.

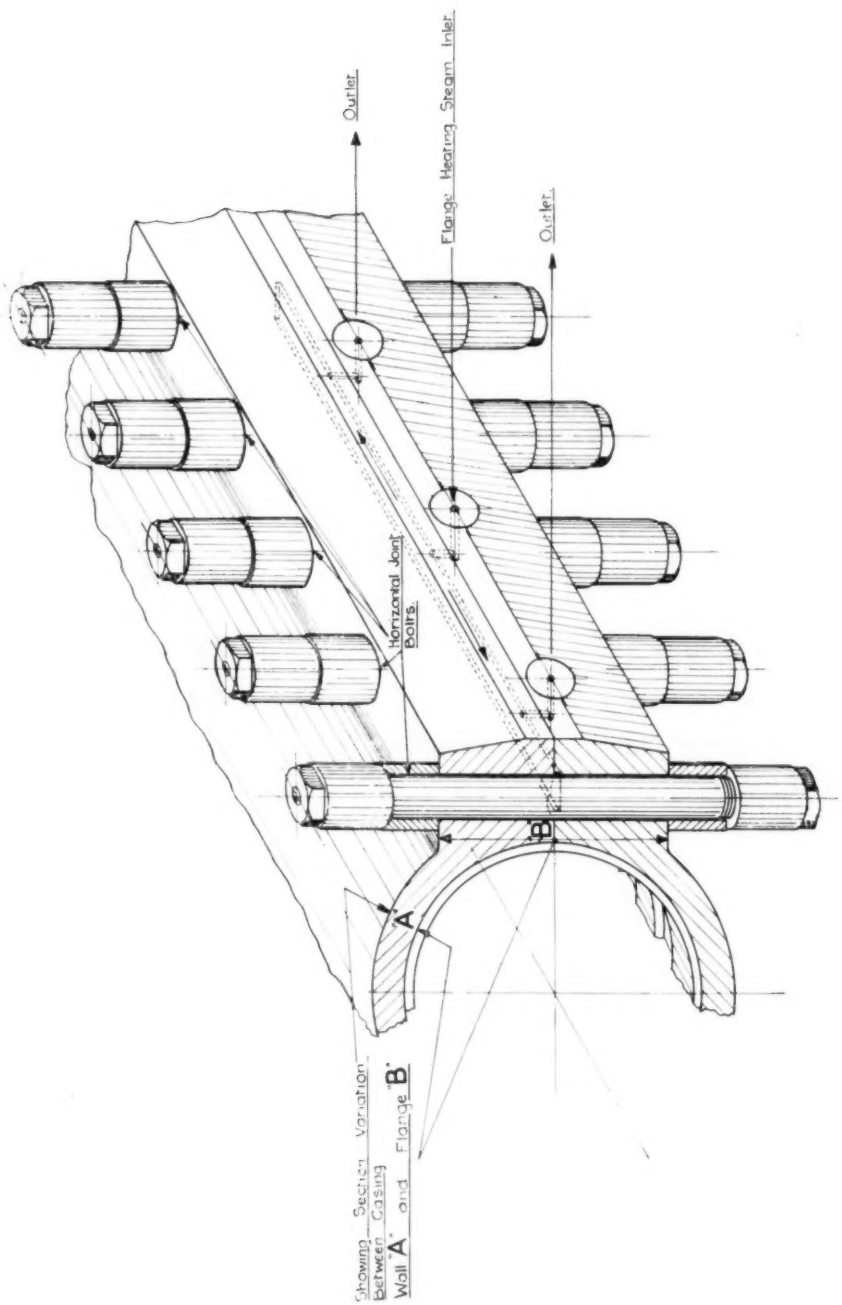


Fig. 7.—Sketch of high pressure casing, showing flange heating arrangement and section comparison

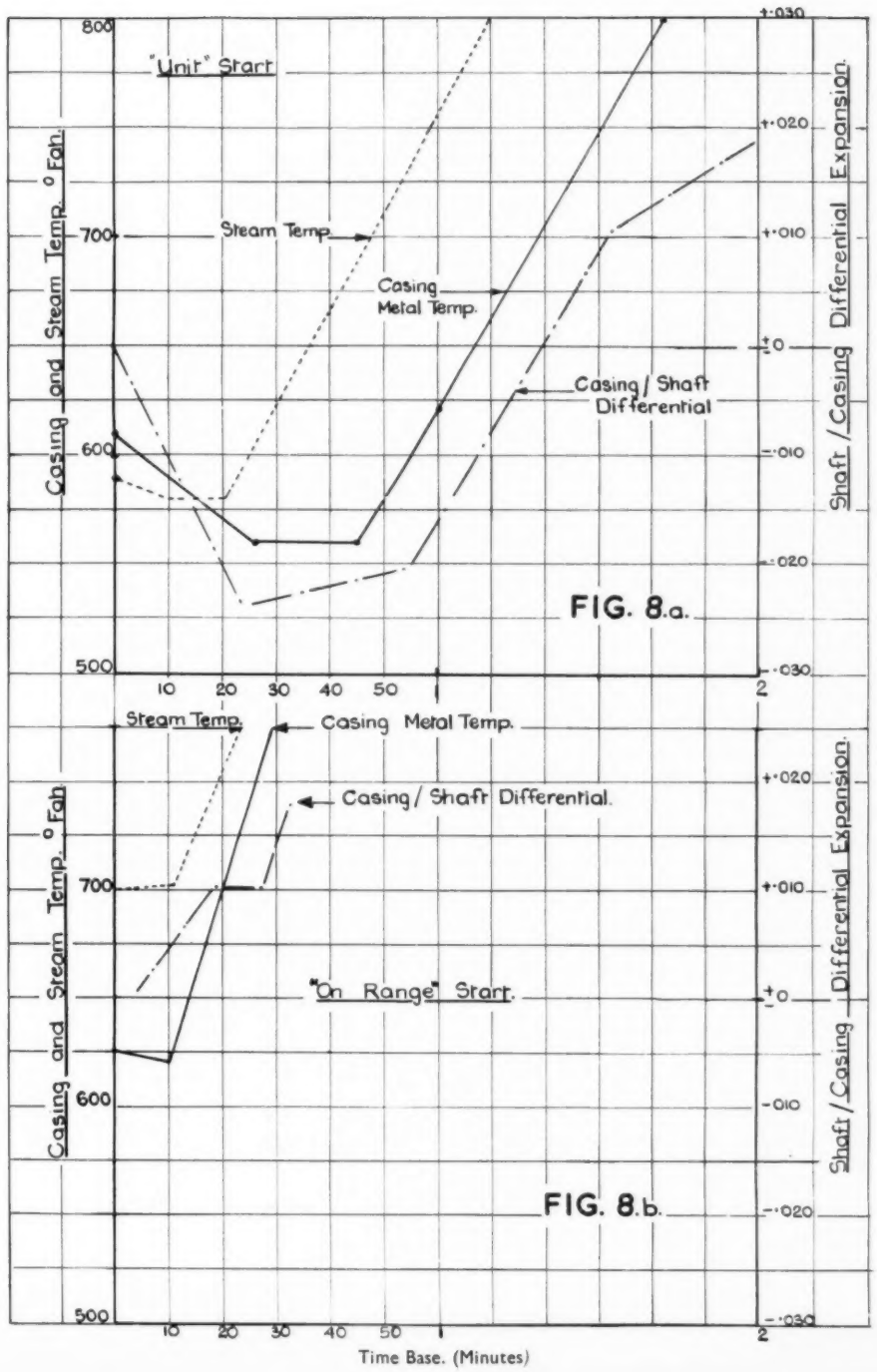


Fig. 8.—Comparison of shaft/casing differential expansion for (a) a 'unit' installation, and (b) an 'on range' installation, with different initial steam conditions

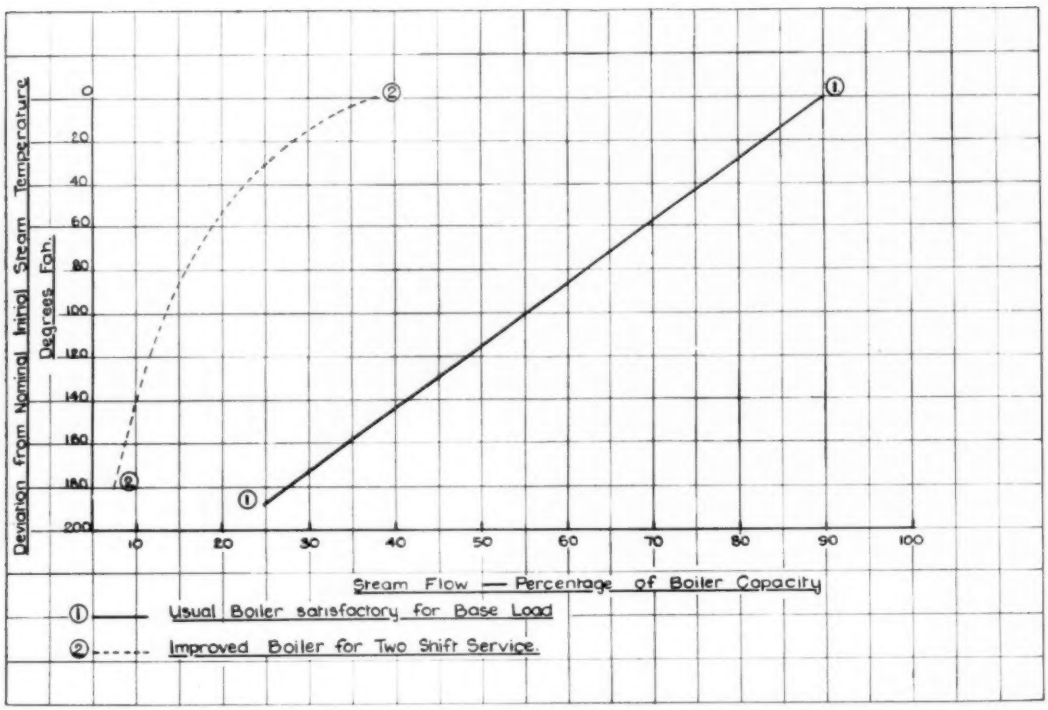


Fig. 9.—Relationship between boiler steam flow and initial steam temperature

It follows, therefore, that a cold turbine is often supplied with superheated steam at a temperature lower than that normally obtained from an 'on range' installation at turbine start-up, and the temperature can then only be increased with the steam flow, speed, and load rise of the machine.

When a machine has been shut down for some 4 to 8 hours, continuous barring is adopted to prevent any rotor distortion due to unequal heating which would occur if left standing. On starting up, the mean casing temperature will be approximately 100° to 200°F lower than when running at full load, and therefore the admission of steam at a temperature lower than the casing and shaft metal temperatures results in a cooling of both components, the amount of cooling being determined by the value of the steam-to-metal differential temperature at starting, and throughout the run-up period. This effect may be seen in Figs. 5 and 8a.

Instrumentation

Appreciation of these factors, and the development of correct operating techniques, has led to the introduction of turbine supervisory instrumentation, which was the subject of a previous article in this Journal.* The extent of this instrumentation covers all the following features :—

1. *Shaft position* or differential expansion values between shaft and casing for both H.P. and I.P. turbines in a 3-cylinder tandem unit. This measurement is indicated optically (Fig. 10) and is also indicated and recorded by electronic means.

*Higgins L. J., and Carson P. F. 'Supervisory aids in high pressure high temperature steam turbine operation.' English Electric Journal, June 1952, Vol. 12 No. 7.

2. *Shaft eccentricity* for both H.P. and I.P. shafts, indicated and recorded electronically.

3. *Casing forward expansion*, indicated and recorded electronically.

4. *Bearing pedestal vibration*, indicated and recorded electronically.

5. *Steam to metal differential temperature*. This measurement is recorded by a two-pen recorder which has the advantage of indicating at any time the differential value and also the temperature level of the two components.

Operating Results

Before any attempt to reduce the time required from starting to carrying full load on any machine, test instrumentation is fitted and data obtained covering the points already discussed.

A brief account follows of a typical test carried out in collaboration with the London Division of the British Electricity Authority on a 3-cylinder tandem 60 MW turbo-alternator set (Fig. 11) designed for operating under steam conditions of 900 p.s.i.g. and 900°F. Owing to boiler limitations, the machine, during these tests, was operating at 52 MW with steam conditions of 600 p.s.i.g. 800°F.

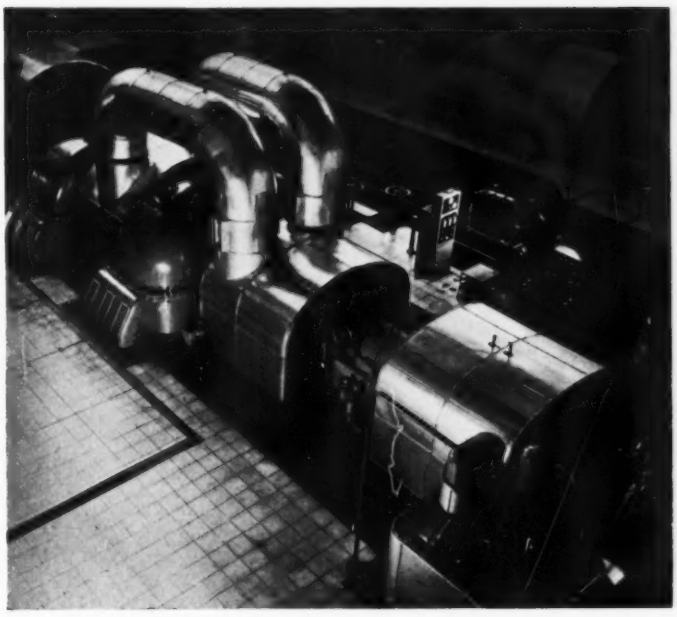


Fig. 11.—*A* 60 *MW turbo-alternator set in the Fulham Generating Station of the London Division of the British Electricity Authority, operating at* 600 *p.s.i.g.* 800°*F, on which the test results given in Figs.* 12 *and* 13 *were obtained*

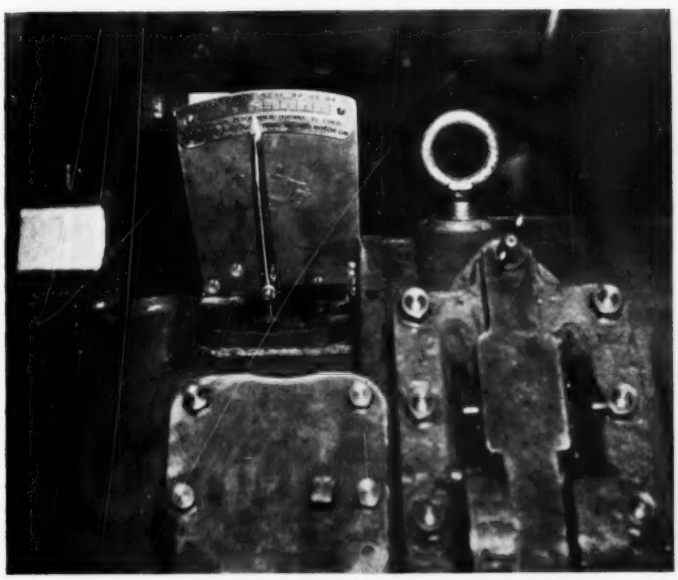


Fig. 10.—*Mechanical shaft-position indicator for use as a datum comparison with electric equipment*

Preliminary data obtained on start-up following an 8-hour shut-down.

This run-up provided information applying to the first test requirement which was a start-up following a shut-down of six to eight hours.

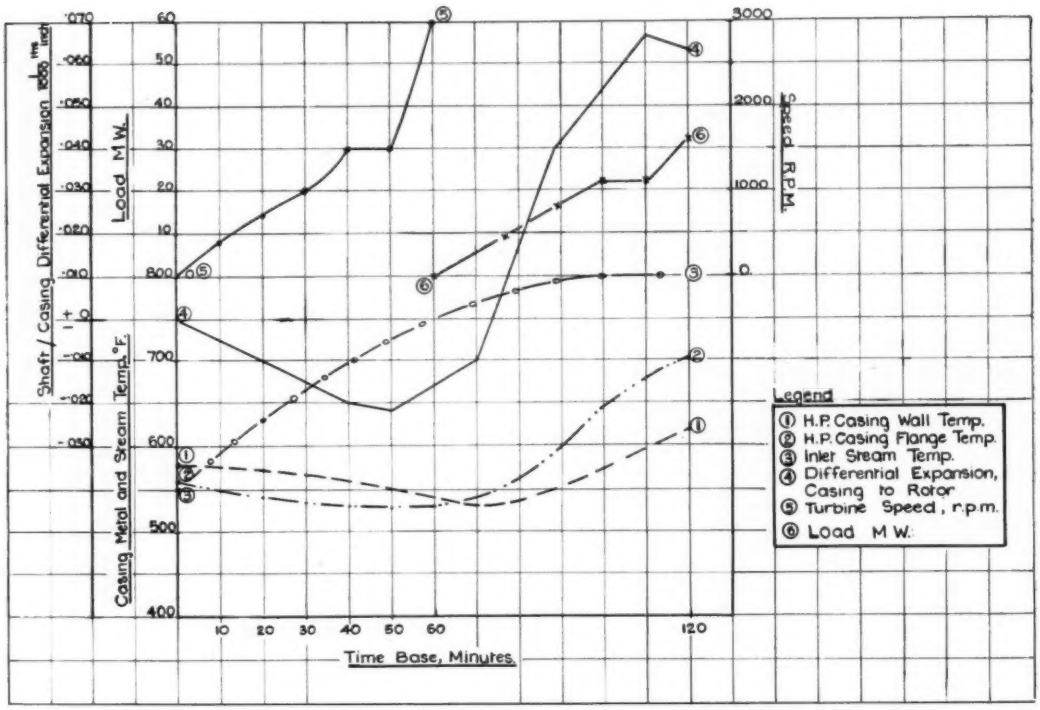

Shaft / Casing Differential Expansion 1000 ins
Load M.W.
Casing Metal and Steam Temps. °F.
Speed R.P.M.
Time Base, Minutes
Legend
① H.P. Casing Wall Temp.
② H.P. Casing Flange Temp.
③ Inlet Steam Temp.
④ Differential Expansion, Casing to Rotor
⑤ Turbine Speed, r.p.m.
⑥ Load M.W.

Fig. 12.—*Data obtained on trial run-up of* 60 *MW turbo-alternator set* (*Fig.* 11) *following a shut-down of eight hours*

The maximum H.P. casing temperature recorded on full load was 750°F, and after a shut-down of eight hours on continuous barring this temperature fell 165° giving a surface metal temperature of 580°F at the inlet belt.

The steam temperature before the turbine stop valve and immediately before steam admission to the turbine was 550°F, giving a fair temperature balance for such starting conditions.

The casing temperature was generally reduced following steam admission, falling steadily throughout the rolling period at a speed of approximately 1,000 r.p.m. to a value of 530°F. On accelerating to 3,000 r.p.m. and throughout further running, a steady and continuous rise in temperature took place. The flange temperatures lagged behind the top inlet belt temperatures by some 80° after 120 minutes steaming.

Associated with this fall in casing metal temperature was a contraction of the casing of 0.025″ maximum and a continuous increase in negative differential between rotor and casing throughout the whole of the rolling period, which occupied 50 minutes of the 60 minutes taken in bringing the unit up to 3,000 r.p.m. This negative differential appeared to be a condition of shaft cooling and reached a value of approximately 0.022″, thereby making adjustment of the backing-off mechanism necessary to maintain sufficient forward axial clearances between moving blades and fixed nozzles.

Upon accelerating from 1,500 r.p.m. to 3,000 r.p.m. an immediate change in direction occurred and a positive differential trend continued until the machine was carrying a load of 30 MW when the expansion of the casing equalled the rotor expansion and consequently reduced the amount of positive differential expansion between the two components.

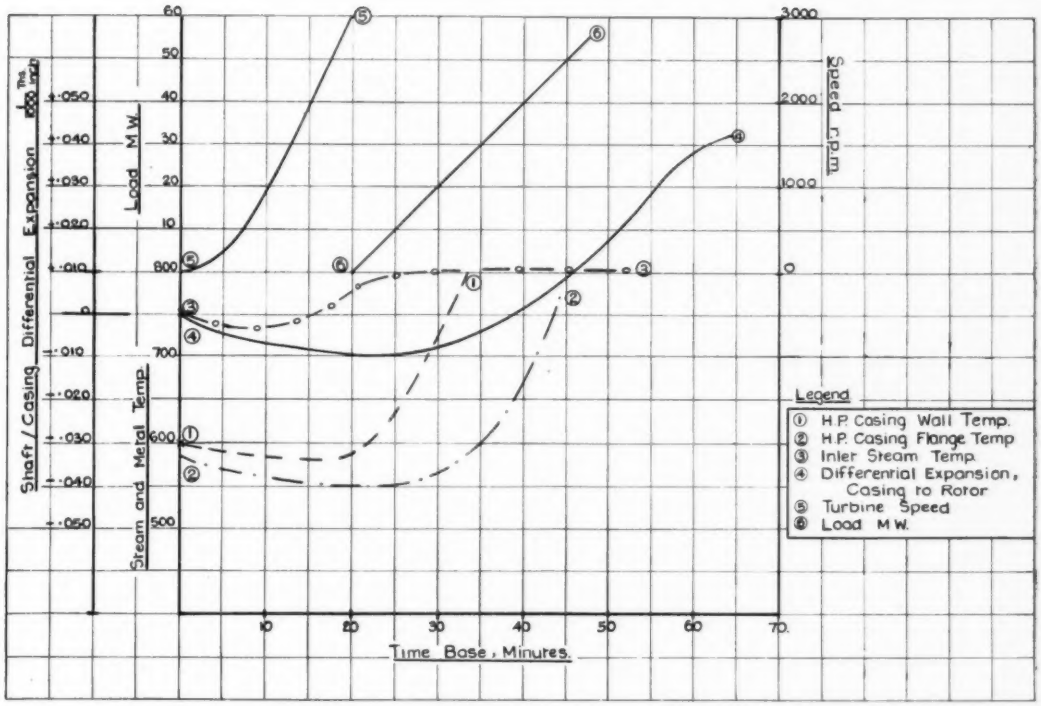


Fig. 13.—*Data obtained on first quick-starting test of* 60 *MW turbo-alternator set* (*Fig.* 11) *following a shut-down of eight hours*

From the data obtained, the maximum positive differential value was of the order of 0.070″ which means that the axial clearance between moving blades and fixed nozzles was increased by that amount.

It was established that the temperature of the gland steam and the steam supply to the flange heating was generally 100°F below metal temperature and had the effect of cooling the flange and also causing a slight increase in eccentricity of the shaft.

The I.P. casing temperature showed little change during the run-up period, there being a fall in temperature of approximately 10°F following the acceleration to 3,000 r.p.m.

With a condenser vacuum of 26″ Hg the L.P. cylinder metal temperature was constant at 125°F during the run-up period. This temperature diminished following load application and it was concluded that 'windage' effect, combined with the relatively low vacuum, assisted in maintaining the I.P. casing temperature.

I.P. turbine casing to rotor differential readings indicated a cooling of the rotors on steam admission with little change in casing expansion values, but a sudden change in differential occurred at 1,500 r.p.m. and a positive differential trend continued up to full load with a maximum value of 0.035″.

The loading rate was .43 MW per minute and the machine was on full load in two hours after reaching synchronous speed, the overall time from commencing running to reaching full load being three hours.

Having analysed all the data obtained during this run and graphically indicated in Fig. 12, a

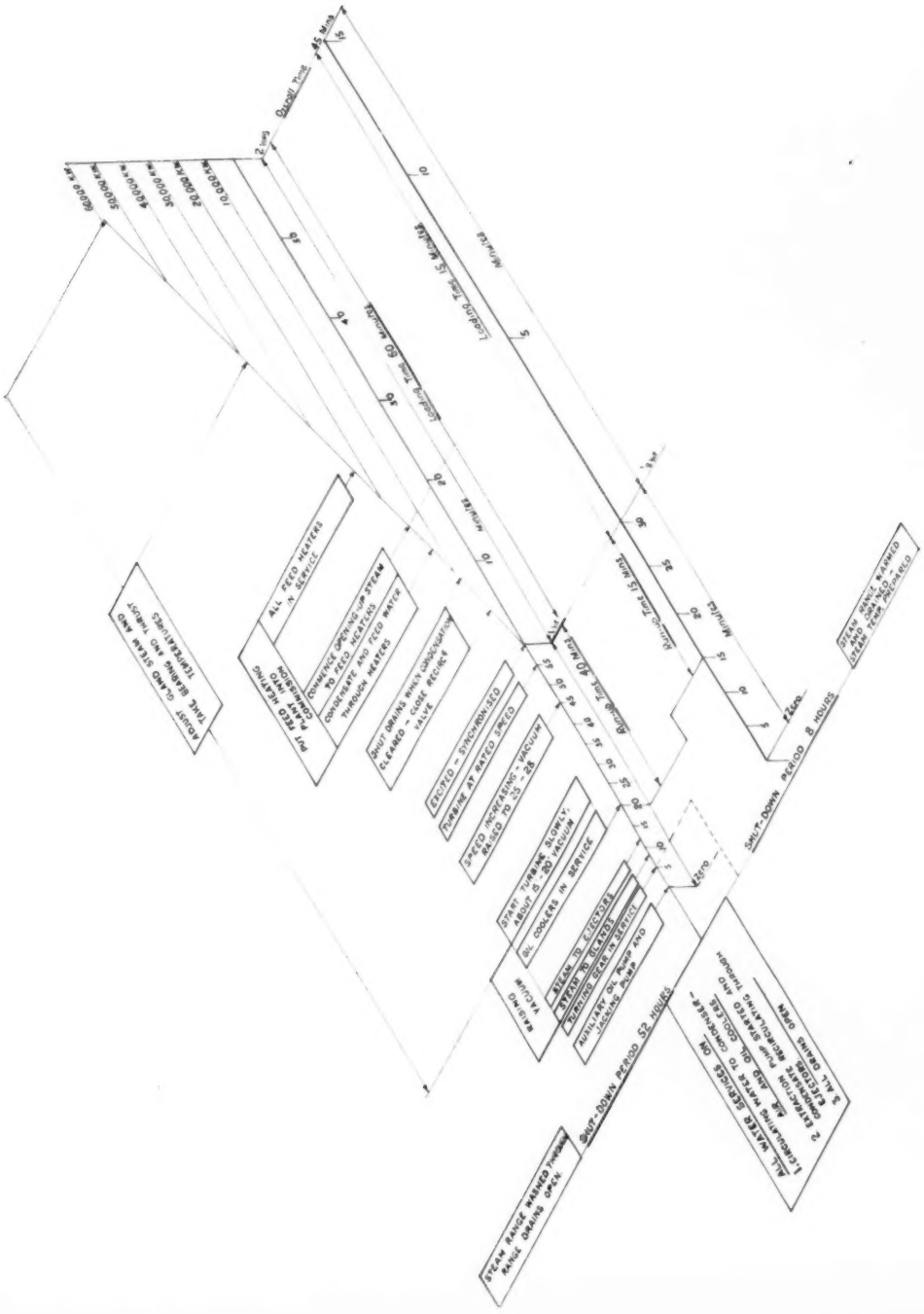

Minutes
Loading Time 15 Minutes
Loading Time 60 Minutes
Loading Time 90 Minutes
Loading Time 10 Minutes
Shut-down Period 8 Hours
Shut-down Period 52 Hours
ALL FEED HEATERS IN SERVICE
COMMENCE OPENING UP STEAM TO FEED HEATERS
CIRCULATE FEED WATER THROUGH HEATERS
SHUT DRAINS WHEN CONDENSATION CLEARED - CLOSE VALVE
ERECTED - SYNCHRONISED
TURBINE AT RATED SPEED
SPEED INCREASING - VACUUM RAISED TO 25 - 28
START TURBINE SLOWLY, ABOUT 15 - 20 VACUUM
OIL COOLERS IN SERVICE
STEAM TO ELECTRICAL TURNING GEAR IN SERVICE
AUXILIARY OIL PUMP AND JACKING PUMP
RAISE VACUUM
ALL WATER SERVICES ON CONDENSATE, ALL DRAINS TO CONDENSATE AND ELECTRICITY MADE TO SYNCHRONISE
2 EXTRACTION PUMP STARTED TO CIRCULATE CONDENSATE AND ELECTRICITY MADE TO SYNCHRONISE
3 ALL DRAINS OPEN
STEAM RANGE MARKED THUS RANGE DRAINS OPEN
STEAM RANGE MARKED THUS RANGE DRAINS OPEN DURING TURN ROUND PERIOD
BY FEED HEATING CHAMBER INTO CONDENSER
DRAIN BLANKS AND TURBINE
Overall Time 45 mins
2 hours
10
5
15
20
30
40
1
60
35
25
45
50
55
00
05
10
15
20
25
30
35
40
45
50
55
00
05
10
15
20
25
30
35
40
45
50
55
00
05
10
15
20
25
30
35
40

Fig. 15.—One of four 50 MW turbo-alternator sets operating at 900°F in the Bromborough Generating Station of the British Electricity Authority, where much operational information was obtained

further test run was made with the object of reducing overall operating time. The following results were obtained.

First quick-starting test following 8-hour shut-down.

Since the duration of shut-down was to be 8 hours, the machine was unloaded and tripped within 30 minutes in order to ensure that the casing temperature on starting up would be of the order of 600°F.

With this casing temperature, the steam temperature before the turbine stop valves prior to steam admission to the turbine was regulated to 750°F, and the vacuum at the turbine exhaust was established at 21″ Hg.

The circulating water quantity to the gland steam de-superheater was regulated to give a gland steam temperature of 550°F, and gland steam was applied prior to raising vacuum and in the sequence of H.P., I.P., L.P.

Shaft eccentricity values remained steady at the maximum value of 0.001″ during the starting period, the general level being 0.0005″.

Flange heating steam at 35 p.s.i.g. was admitted almost simultaneously with gland steam application, the temperature differential between casing walls and flanges reaching the maximum of 175°F.

The machine was then brought up to 3,000 r.p.m. in an overall period of 20 minutes.

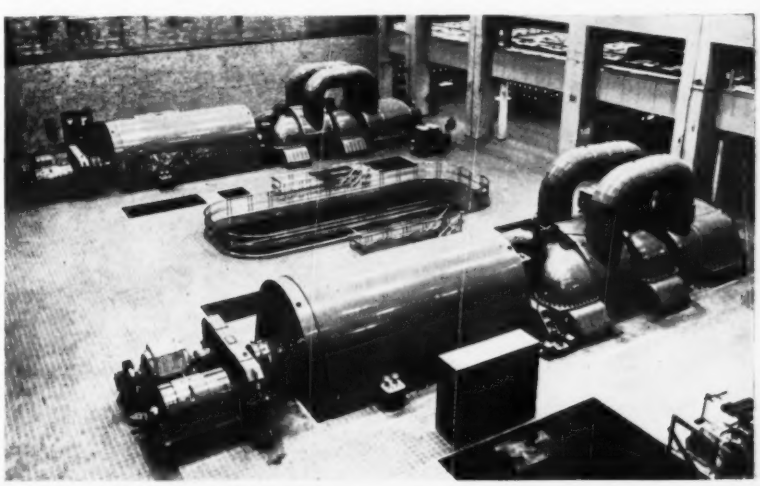


Fig. 16.—Two of four 60 MW 3,600 r.p.m. turbo-alternator sets operating at 850 p.s.i.g. 900°F in the J. Clark Keith Generating Station, Windsor, of the Hydro-Electric Power Commission of Ontario, where considerable operating investigations were carried out

A negative differential value of 0.010" developed between H.P. shaft and casing during this period of running and, the I.P. conditions showing little change, the machine was immediately synchronised and a load of 5 MW applied, following which loading was continued at a steady rate of 2.0 MW per minute. The total time from steam admission to full load was 46 minutes, a considerable improvement over previous operating times.

A graphical indication of the data obtained during this test is given in Fig. 13.

Further tests of a similar nature have brought about even faster running-up times, and a higher rate of loading, and the following table gives operating times achieved in a series of tests :—

Period of shut-down, hours		5	8	15	50
Total run-up time to synchronous speed, minutes		15	20	20	35
Time from synchronous speed to full load, minutes.	30 MW	10	20	30	90
	50 „	13	28	50	120
	60 „	15	30	55	125
Overall time from start to full load, minutes	30 „	25	40	50	125
	50 „	28	48	70	155
	60 „	30	50	75	160
Mean loading rates in MW/minute.	30 „	3	1.5	1	0.33
	50 „	3.75	1.8	1	0.4
	60 „	4	2	1.1	0.5

Note : *Load application following re-synchronising after instantaneous rejection of full load has been at the mean rate of 6 MW/minute.*

Conclusions

From the data obtained and the behaviour of the machines during tests it is believed that the rate at which a steam turbo-alternator set can be run up and loaded is directly controlled by the steam-to-metal temperature difference prevailing during the overall starting and loading cycle. Appreciation of this fact leads to operating techniques which show an appreciable reduction in operating times, together with safe and reliable operation.

During tests, a 60 MW three-cylinder machine has been run up to speed and synchronised in 10 minutes following a shut-down of five hours, and although the loading was then at a mean rate of 4 MW/minute, experiments have been made with loading rates of 10 MW/minute.

It is believed, however, that the time required for the operating engineers to ensure that all aspects of overall plant operation are under control is a deciding factor in the minimum time of run-up and loading. To assist in visualising some of the points involved, Fig. 14 indicates the various items requiring attention when operating turbine plant. A shortening of the overall run-up and loading period must evidently be related to the ability of the operator to exercise adequate supervision of these features, and the simplified control of certain items is in the process of development.

In conclusion, the author would express his appreciation of the facilities and data provided by the British Electricity Authority, the Hydro-Electric Power Commission of Ontario, and the Steam Turbine Design Department of The English Electric Company, in carrying out all the tests associated with this article.

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